POWER

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The Franklin Institute

THE Franklin Institute was founded by far-seeing and public-spirited citizens of Philadelphia in 1824.

Its stated object was the promotion of the useful arts—in Philadelphia.

But it has made for itself a place in the Nation and a name in the World that made the leading exponents of Science and Engineering and Industry glad to cross oceans and continents to have a part in the ceremonies with which its completion of a century of honored existence and distinguished service was signified.

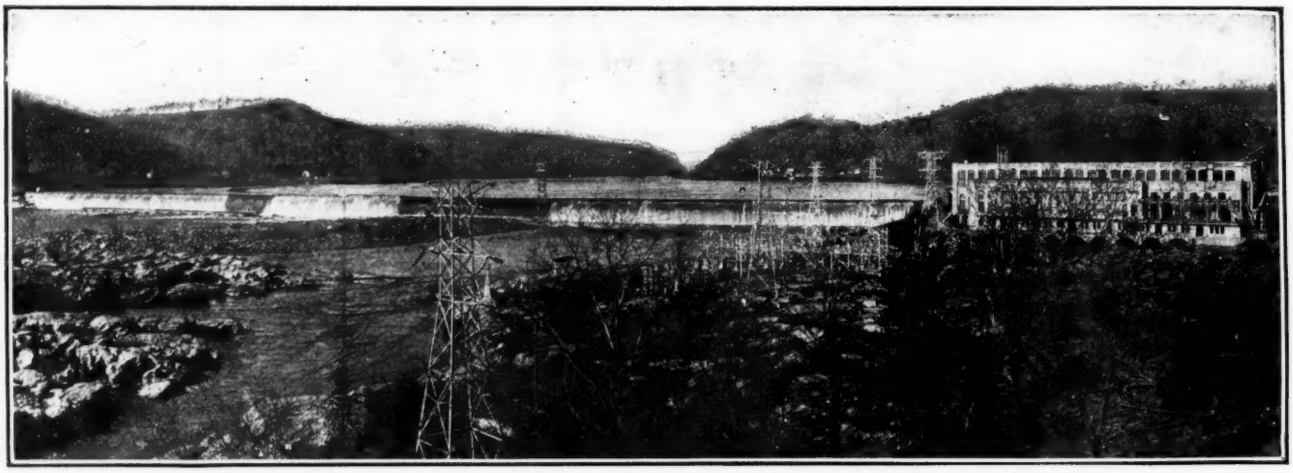
The contributors to its program, enumerated on another page, include the foremost thinkers and workers in the struggle to wrest her secrets from Nature and utilize them in the service of mankind.

How the great statesman, philosopher and scientist in whose name and honor the Institute was founded would have gloried in that occasion! How he would have reveled in association and discourse with the brilliant men that it brought together! How he would have exulted in the development of the science with the beginnings of which his name is inseparably connected, into the great social-economical and industrial factor, the exquisitely delicate but wonderfully potent tool for further probings which it has become, perhaps, as one speaker expressed it, the structural basis of the universe itself!

I wonder—if it is a fact that consciousness and intelligence and a sympathetic interest in the doings of those that one has left behind persists after death—with what feelings and emotions he regarded that gathering; how he looks upon our gropings in subjects nearly as embryonic as electricity was when he became interested in it.

And if prescience is an attribute of his condition, I wonder what marvels he sees ahead for us through the impressment of another century of research, invention and development.

F. R. Low



Holtwood power house and dam, the latter being 2,392 ft. long and 55 ft. high

Fifteen Years' Progress in Water-Power Development at Holtwood*

IN EXTENT of drainage basin and in total and average run-off the Susquehanna is the largest river flowing into the North Atlantic Ocean with the exception of the St. Lawrence. Its fan-shaped watershed includes 27,400 square miles in the states of New York, Pennsylvania and Maryland. The river is unique in that the steepest slopes occur in the lower reaches where the stream has cut its way through the Piedmont Plateau. Tidewater from the Chesapeake Bay extends only five or six miles up the river, whereas in other large streams along the Central Atlantic Coast, such as the Hudson, the Delaware and the Potomac, there are long stretches of navigable water at the river mouths. From Columbia, Pa., to tidewater, a distance of 39 miles, the Susquehanna drops 225 ft., giving an average slope of nearly 5.8 ft. per mile. This condition is advantageous in that the best water-power sites are thus located near the large industrial centers along the Atlantic seaboard and the greater part of the drainage basin is above these sites. The run-off from all but 630 square miles of the total drainage area is available at the Holtwood development of the Pennsylvania Water & Power Company.

The flow of the Susquehanna is subject to extreme fluctuation. The average flow at Holtwood is about 40,000 cu.ft.-sec. The maximum recorded flood, probably about 700,000 cu.ft.-sec., is the so-called "Johnstown Flood" of 1889. This occurred in June and was due entirely to heavy rainfall, although usually the maximum run-off in a year occurs at the time of the spring thaw. The minimum flow of the river, like the maximum, is somewhat uncertain. Flows at McCall Ferry near Holtwood, of 2,620 cu.ft.-sec. in September, 1900, and 2,200 in December, 1909, have been recorded, but these figures are of doubtful value. Since the Holtwood plant went into operation in October, 1910, there has been an opportunity to estimate the discharge more accurately by using the turbines as water meters. The minimum flow since that time was 3,200 cu.ft.-sec. For practical purposes the minimum flow is now taken as 3,000.

The conditions of rapid descent, steep banks and absence of populous towns in a region within easy reach of large industrial centers attracted the attention of water-power investigators to this region early in the history of hydro-electric development. Several schemes were projected by rival promoters and engineers. Finally, a merger of certain interests led to the formation of the McCall Ferry Power Co., which in 1905 began actual construction work on the McCall Ferry, now known as the Holtwood development. This was the largest water-power project of the kind that had ever been attempted up to that time. This company failed in 1907, and in 1910 the Pennsylvania Water & Power Co. was organized and took over the work on the development. The first unit, rated at 13,500 hp. under a head of 53 ft., went into commercial operation in October, 1910.

The power-house substructure, being practically a part of the dam, was of necessity built to its ultimate size at the start, and the number of main units originally decided upon could not be changed. However, the proposed capacity of the

In October, 1910, the first unit was put into operation in the Holtwood plant of the Pennsylvania Water & Power Company at Holtwood, Pa., on the Susquehanna River. The first five units were of the double runner vertical-shaft type, discharging into bent draft tubes and rated at 13,500 hp. under 53-ft. head. Units 6 and 7 are the same as Nos. 1 to 5 except that they are rated at 17,000 hp. under a 62-ft. head. Unit No. 8 is a single-runner vertical-type machine rated at 16,500 hp. under a 63-ft. head and discharges into a bent draft tube. The plant has recently been completed by the installation of units 9 and 10, rated at 20,000 hp. under a 62-ft. head and discharging into spreading draft tubes of special design, with a scroll case at the lower end similar to that for the runner setting. The horsepower of the units has been increased about 25 per cent, and the efficiency increased from 83 to about 92 per cent.

*This article is based on the three papers describing the Holtwood installation, by N. B. Higgins and R. L. Thomas of the Pennsylvania Water & Power Co.; Frank H. Rogers, Wm. Cramp & Sons Ship and Eng. Bldg. Co.; and J. Walter May and K. D. MacLean, Day & Zimmermann, Inc. These papers were presented at a joint meeting of the Baltimore Sections of the A.I.E.E. and A.S.M.E. at Holtwood, March 1, 1924.

units has been increased and the form of the water passages much altered. The first section of the powerhouse superstructure was built for five units and the exciter bay. The next section housed units Nos. 6 and 7 and the next No. 8. Contracts for building the final extension for units Nos. 9 and 10 were let in March, 1923. Prior to 1923 all energy was generated and transmitted at 25 cycles. This was the generating frequency in both Baltimore and Lancaster before the

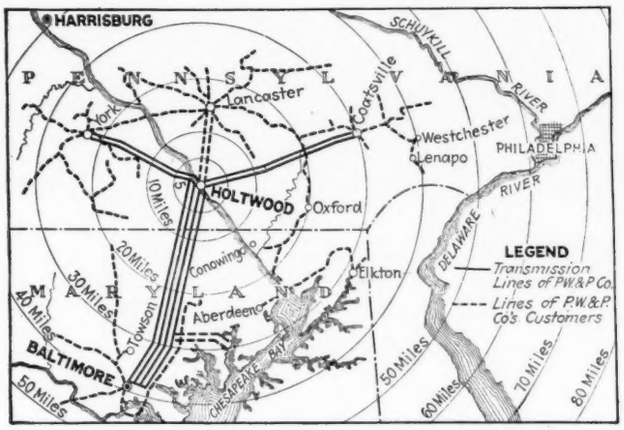


Fig. 1—Location of Holtwood with relation to surrounding country

Holtwood plant went into operation. All other neighboring systems, however, are 60-cycle, and the 60-cycle load in both Baltimore and Lancaster is growing much more rapidly than the 25-cycle load. These considerations, and especially the securing of contracts to supply energy to the Edison Light & Power Co. of York and the Chester Valley Electric Co. of Coatesville, led to the adoption of 60 cycles for the 1923 extension. Fig. 1 shows the location of the Holtwood development and the connecting transmission lines. At Conowingo, about 13 miles below Holtwood, is the site of a second large development which is much talked about at the present time.

The dam is of the gravity ogee type. Its length from abutment to abutment, including the log chute, is 2,392 ft. The clear spillway length is 2,368 ft. with an average height of 55 ft. In low-flow periods flashboards are placed on the dam, the boards now used consisting of panels 3 ft. high and 16 ft. long made of 1½-in. Georgia pine supported by pins made of 2½-in. standard pipe.

Installing Modern-Type Units

The decision to install modern-type single-runner turbines with their spiral scroll cases and spreading draft tubes also of spiral formation, rendered the substructure of the old-type units unfit for the installation of the new turbines. The outline of the new draft tubes with the line of the old work superimposed is shown in Fig. 6. To meet the requirements for the new turbines, the old substructure had to be unwatered, and 2,700 cu.yd. of old concrete and rock removed and new concrete placed to accommodate the draft tubes, scroll cases and wheel pits in conformity with the design for the new 20,000-hp. units.

In addition to putting in a new substructure for the two turbines, the construction work involved a concrete superstructure over the generator room 100x58x57 ft., a concrete superstructure 160x32x75 ft. for the transformer house, which was also to house the two frequency changers, and a concrete superstructure 110x63x44 ft. for the gate house.

Construction work was started on the wheel pits in May, 1923, by Day & Zimmermann, Inc., and the first unit was turned over the latter part of January of this year. It is worthy of note that the work was carried to completion without a single casualty, although at times over 800 men were employed on the job. A feature that materially assisted in making it possible to complete the work in so short a time was the building of a large part of the concrete forms in sections previous to their being used, so that when the excavation was completed these forms were ready to be placed with a minimum delay.

Advance in Design of Hydraulic Turbines

The great advances made in the last fifteen or twenty years in the design and construction of hydraulic turbines is illustrated in a striking manner in the units installed in the Holtwood station. This station might readily be considered a testing laboratory on a large scale, for testing draft tubes, as three different types of tubes have been installed. Fig. 3 shows a cross-section through unit No. 3 and Fig. 4 a cross-section through unit No. 8. Unit No. 3 is rated at 13,500 hp. under a 53-ft. head at a speed of 94 r.p.m. The first seven units installed in this plant are of this same type, having two runners mounted on a vertical shaft, each runner discharging into an elbow-type draft tube. Owing to the sharp curvature of these tubes they are inefficient, regaining but a small part of the energy discharged from the runners. Unit No. 8, Fig. 4, is rated at 16,500 hp. under a head of 63 ft. at a speed of 94 r.p.m. This unit is of the single-runner type, with volute casing formed

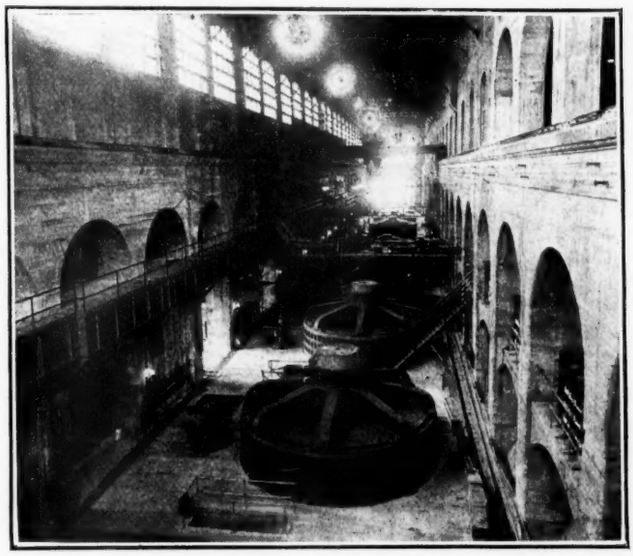


Fig. 2—Ten units installed having a total capacity of 158,000 horsepower

in the concrete. The curvature of the draft tube is much less severe than in the case of the older units. At the time this unit was installed, it was appreciated that the draft tubes and settings of the older units were inefficient and therefore considerable money was spent to cut out and change the existing concrete substructure, to permit the installation of the single-runner type unit. The actual efficiencies developed by this unit as compared with the older units, more than justified the additional expense.

It might also be pointed out that in the case of unit No. 8 all parts of the operating gear with the exception of the guide vanes are located in the turbine operating pit, where they are available for lubrication, inspection and repairs, while in the case of the older units all the operating gear is submerged.

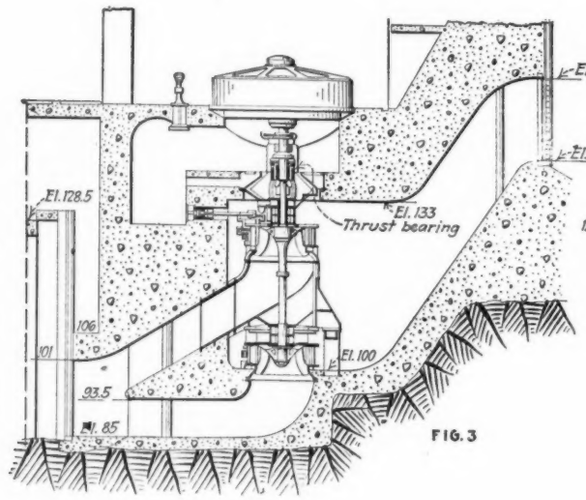


FIG. 3

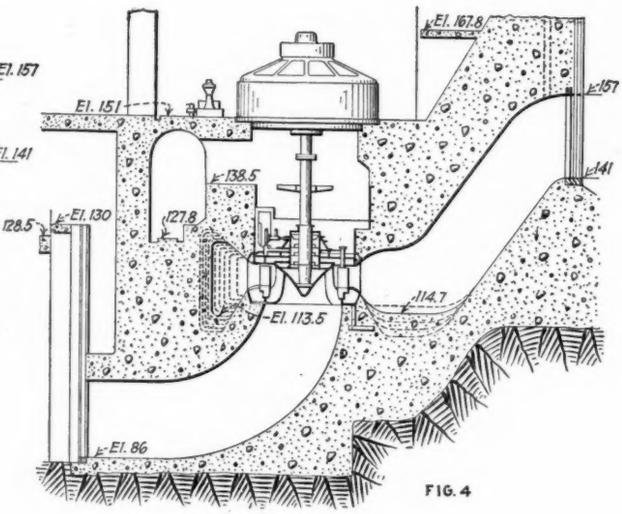
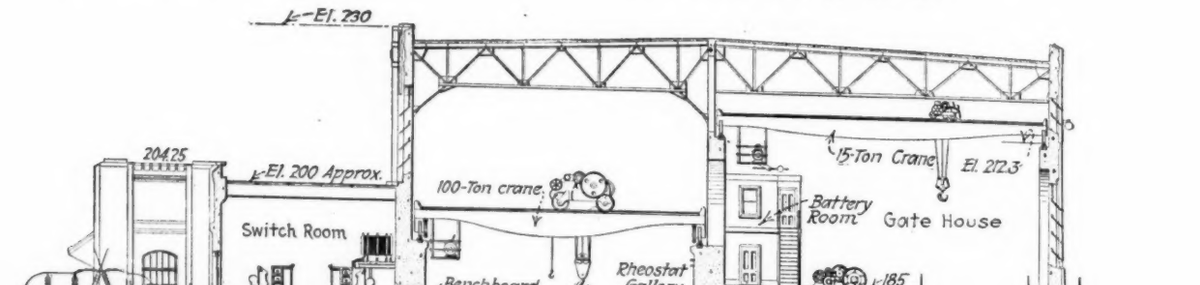


FIG. 4



Figs. 3 to 5—Sections through the Holtwood power house

Fig. 3—Type of construction used for units 1 to 7. Fig. 4—Section through unit 8. Fig. 5—Cross-section through new extension to plant which contains units 9 and 10.

In Fig. 5 is shown a cross-section through one of the new units, two of which have been recently put into commercial operation. Each of these units is rated at 20,000 hp. under a head of 62 ft. at a speed of 94.7 r.p.m. Their capacity is about 25 per cent greater than that of the former machines, when referred to the same head, but in spite of this increase in power the speed is maintained the same as for the older units. For this reason considerably higher specific-speed runners were required for the last two machines, and as the intake gates and tailrace piers were already in place, it was necessary to prepare special designs of casings and draft tubes to suit the restricted space. These units are of the vertical-shaft single-runner type as in the case of unit No. 8, provided with volute casing molded in the concrete, and draft tube of the Moody type with a spiral collector.

The runners are of the Francis type, having a specific speed of 84.4 (English units), and are of cast iron containing sixteen vanes, each cast integrally with the crown and band. The throat diameter of the runner is 149.5 in., the maximum over-all diameter is 13 ft. 8 in. and the weight 88,600 lb. The speed ring is made in two sections and consists of ten vanes cast integrally with

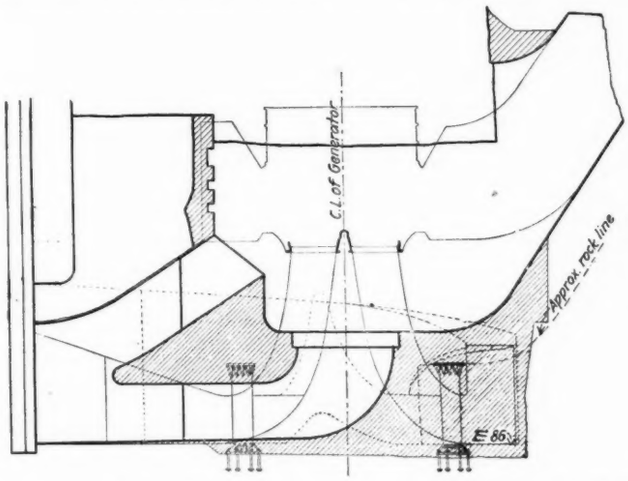


Fig. 6—Heavy lines show outline of old concrete that had to be removed to make way for the new setting, shown in light lines

the upper and lower crown plates. These vanes are designed to carry the total weight of the generator and turbine and a portion of the concrete between the generator and casing, amounting to 3,865,000 lb. The vanes are also designed for the maximum upward hydraulic pressure that could occur when 100 per cent load is suddenly dropped. The lignum-vitæ bearing is of the adjustable type, made in eight radial sections, each guided in machined tapered grooves.

Draft Tube with Spiral Collector

The design of draft tubes installed with these units is shown in Fig. 7. In this particular installation the space allowed for installing these tubes was restricted and it was therefore not possible to flare out the bell of the tube to the diameter desired at the bottom to obtain low velocities of discharge and low components of whirl. It was therefore necessary to provide a spiral collector chamber around the bottom of the draft tube bell, so that the relatively high velocities at this point could be efficiently regained. Owing to the location selected for the baffle in the collector passage, about three-fifths of the total flow is discharged through the right-hand opening at the tailrace and the remainder through the left-hand opening. Another feature in the design of the draft tubes are the seven cast-iron columns under their lower ends. These columns are similar in form to the stay vanes in the scroll cases and are placed so as to assist in turning the outflowing water toward the tailrace.

A comparison between the energies to be regained in the draft tubes in the different units in this station is shown in Fig. 8. The upper curve shows specific speed plotted against the absolute velocity head at the outer portion of the discharge from the runner, in per cent of the total effective head. For units Nos. 1 to 5 this velocity head amounts to 17½ per cent of the total head; for unit No. 8, to 30 per cent of the total head; and for units Nos. 9 and 10, to nearly 45 per cent of the total head. As the various units have been installed in this plant, it is evident therefore that the function of the draft tube has become more important, and in order to indicate the relative importance of draft-tube efficiencies in the case of the three types of units considered, a second curve has been drawn through points of assumed draft-tube efficiency, to show the amount of energy regained by the use of the various types of draft tubes.

Improvements in Efficiency Due to Draft Tube

The theoretical increases in efficiencies possible for the older units if equipped with the modern draft tubes are indicated by the light curves drawn through the 30, 60 and 80 per cent efficiency points. It must not be assumed that the possible gain in efficiencies for units Nos. 1 to 5 and unit No. 8 could be directly read from these curves, for it must be remembered that the velocity-head curves are all calculated on the basis of the axial velocities plus the whirl velocities at the outer portion of the runner discharge, at which point the whirl is of the greatest magnitude. These curves, however, do indicate that a considerable gain in efficiency would result in the older units if they could be provided with tubes of the same type as on units 9 and 10. A comparison of the tests made on units 2 and 8, shown in Fig. 9, shows a gain in efficiency for the latter of 8½ per cent, due principally to the improvement in draft-tube design. In this same figure are shown the expected results from units Nos. 9 and 10, based on the laboratory tests made on a model of this runner and draft tube. It should be borne in mind, in comparing these various curves, that the runner installed with units Nos. 9 and 10 is of considerably higher specific speed than the runner in unit No. 8, and has very much higher specific speed than the

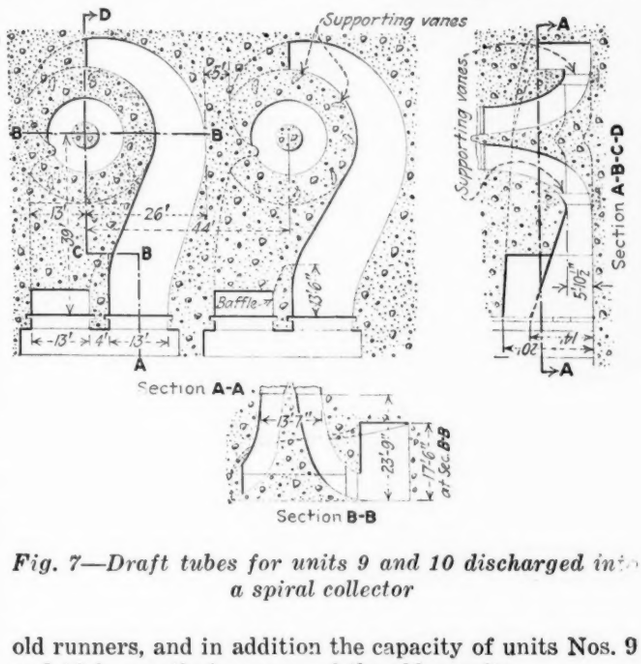


Fig. 7—Draft tubes for units 9 and 10 discharged into a spiral collector

old runners, and in addition the capacity of units Nos. 9 and 10 is greatly in excess of the older units.

The two new generators are 60-cycle machines, whereas the older ones are 25-cycle and the two systems are tied together with two 5,000-kw. frequency-changer sets. All are three-phase, star connected, but the neutrals are not grounded. The first three generators installed were part of an order placed by the McCall Ferry Power Co. and were rated at only 7,500 kw. at 90 per

cent power factor, although the turbines were rated at 13,500 hp. at 53-ft. head. This meant that the turbines could not be operated at full capacity, and it soon became apparent that the loss in available energy in high flow

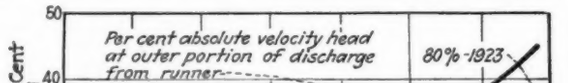


Fig. 8—A comparison between the energy to be regained in the draft tubes on the different units, based on 54-ft. head

would more than pay for the cost of larger generators. Units Nos. 1 and 2 were rewound soon after installation; No. 3 was not rewound until 1923, and in the meantime was operated at 100 per cent power factor at a temperature rating to utilize as much of the turbine capacity as possible. It may be laid down as a general principle that in a plant such as this the generators should be large enough to allow the turbines to operate at maximum capacity in high flow. At such times hydraulic efficiency is of no moment (except as additional turbine discharge may cause higher tail-water) as water is wasting over the dam and the desideratum is to get maximum output.

Rewinding of Generator Units

The rewinding of No. 3 generator well illustrates the progress that has been made in generator design and construction during the life of the Holtwood power house, as an 8,333-kva. winding was replaced with one rated at 13,750 kva. without change in the armature iron or field coils. The old armature winding, consisting of round wire, was replaced with coils of rectangular wire. The old winding was insulated with mica and varnished cloth, while the new winding is insulated with all mica. The shaft and coupling of the generator were large enough to transmit the increased output.

A small feature of design which, nevertheless, may save much time and expense in repair work, is the supporting of the lower bearing bracket for the rotor on the bedplate itself rather than on flanges on the stator frame. Seven of the ten Holtwood generators are built in this way. This construction permits the stator and

TABLE OF IMPORTANT DATA, HOLTWOOD PLANT, PENNSYLVANIA WATER & POWER COMPANY

Location of plant	Holtwood, Pa.
Extension made in 1923	40,000 hp
Designed and constructed in accordance with the power company's general plans	Day & Zimmermann, Inc.
Number of units	10
Present capacity, hp.	158,000
Ultimate capacity, hp.	158,000
Width of power house, outside, ft. at transformer house	173 ft.
Length of power house, outside	510 ft.
Width of transformer house	54 ft.
Width of gate house	63 ft. 8 in.
Width of generator room	55 ft. 4 in.
Distance from bottom of draft tubes to highest point of roof coping	144 ft.
Length of dam between abutments includes log chute	2,392 ft.
Length of spillway	2,368 ft.
Average height of dam	55 ft.
Width of dam at base	65 ft.
Dam designed for 17.5 overflow without tension at heel and without vacuum under nappe	
Concrete in dam, cu.yd.	200,000
River's drainage above Holtwood, sq.miles	26,800
Maximum recorded flow, cu.ft.-sec.	700,000
Minimum flow, cu.ft.-sec. (approximately)	3,000
Flow available 50 per cent of the time, cu.ft.-sec.	25,000
Full-load discharge of 10 Holtwood units, cu.ft.-sec.	31,500

FULL-LOAD WATER VELOCITIES, FEET PER SECOND, WITH FOREBAY ELEVATION AT 165

Units	Racks	Penstock Gates
1 to 5	4.3	8.3
6 and 7	4.1	7.9
8	3.8	7.3
9 and 10	4.5	8.6

PENSTOCKS AND GATE HOUSE

Curtain wall extends below elevation of dam crest, ft. 5
Racks, five sections high, made of $\frac{3}{8}$ or $\frac{7}{16}$ x 2 or 2½ in.
Racks inclined with a batter of 1 to 4
Headgates at penstock mouth, roller type of special design made of steel with wooden bumpers and sills. They are raised and lowered from a motor-driven shaft.
Size 16 ft. 6 in. by 6 ft. 11 in.
Number per turbine 4

TURBINES GOVERNORS AND DRAFT TUBES

Data on Old and New Units:	Units 1 to 5	Units 6 to 7	Unit 8	Units 9 to 10
Rated horsepower	13,500	17,000	16,500	20,000
Rated head, ft.	53	62	63 (net)	62 (net)
Rated speed, r.p.m.	94	116	94	94.7
Number of runners	2	2	1	1
Dia. of runners at distributor, in., center line	100	97.5	117.9	120.75
Efficiency, per cent	83	85	91.5	92.5
Dia. of runners at throat, in.	111.5	109.0	147	149.5
Elevation of distributor, centerline:				
Upper runner*	125.95	125.95	121.5	116.0
Lower runner	102.95	102.95	.....	.....
Spec. speed at rating (Eng. units)	54.0	61.5	68.0	77.0
Spec. speed at 54 ft.	53.5	65.9	73.5	82.4
Horsepower under 1 ft. head	34.92	34.79	32.98	42.83
Horsepower under 54 ft. head	13,880	13,810	13,090	17,000
Weight of revolving element, lb.	110,000	125,000	117,000	140,000
Water thrust at 54-ft. head, lb.	75,000	75,000	95,000	115,000
Water thrust at 63-ft. head	90,000	90,000	110,000	145,000

*Standing tailwater elevation with power house shut down and no flow over dam, 101.8; normal full-load elevation about 118. Head varies from 53 to 63 ft.

Manufacturers of turbines and governors	Wm. Cramp & Sons Ship & Eng. Bldg. Co.
Draft Tubes for Units 9 and 10:	
Type	Moody spreading with spiral discharge
Bottom of runner above bottom draft tube	24 ft. 3 in.
Diameter at runner discharge	13 ft. 7 in.
Length of bell to runner	18 ft. 6 in.
Height of cone; extends to runner hub.	
Size of discharge to tailrace, two openings 13 ft. wide by 20 ft. and 14 ft. high.	
Governors, Units 9 and 10:	
Manufacturer	Wm. Cramp & Sons Ship & Eng. Bldg. Co.
Type. . . . . I. P. Morris, double floating lever, belt driven from mainshaft. Governor fluid, water under 275 lb. pressure supplied from central pumping system.	
Capacity of governor, ft.-lb.	190,000

MAIN GENERATORS, UNITS 9 AND 10

Manufacturer. . . . . Westinghouse Elec. & Mfg. Co.
Size 15,000 kva., 0.80 power factor, 13,200 volts, 3-phase, 60 cycles
Guaranteed efficiency, full load, 95.3 per cent; ¾ load, 94.8 per cent; ½ load, 93.6 per cent, at 80 per cent power factor.
Number of poles, 76; speed 94.7 r.p.m.
Exciters, two waterwheel driven, 500 kw., 250 volts, and storage battery (part of old equipment) and two motor-generator sets.
Thrust bearings, Kingsbury type, mounted on top of generators, designed for 543,000-lb. load.

MAIN TRANSFORMERS (1923, 60-CYCLE EXTENSION)

Manufacturer, four 3 phase 60-cycle, 5,000-kva. . . . . General Electric Co.
Type, self-cooled tubular tanks
Manufacturer, three single-phase 60-cycle, 2,000-kva. . . . . Pittsburgh Transformer Mfg. Co.
Type, self-cooled corrugated tanks.

FREQUENCY-CHANGER SETS

Manufacturer. . . . . General Electric Co.
Number. 2
Size, 60-cycle end, 7,140 kva. at 0.70 power factor 13,200 volts, 24 poles. Will carry 6,000 kva. as synchronous condenser. Size, 25-cycle end, 6,000 kva. at 0.90 power factor 11,000 volts, 10 poles.

upper bearing bracket to be removed with the rotor remaining in place and supported on the lower bearing bracket, and it permits the upper bearing bracket, the rotor and the lower bearing bracket to be removed with the stator remaining in place.

Generators Nos. 9 and 10 are guaranteed to give 15,000 kva. at 80 per cent power factor with a temperature rise of 60 deg. C. in the armature as measured by temperature coils and 80 deg. C. in the field as measured by resistance. The outside diameter of the foundation ring is 26 ft. 8 in., and the flywheel effect is 16,200,000 lb. An overspeed of 100 per cent is provided for in the machines, and the guaranteed efficiency at full load is 95.3 per cent, three-quarters load 94.8 per cent, and one-half load 93.6 per cent, all at 80 per cent power factor.

There are eight brake jacks, which are supported by

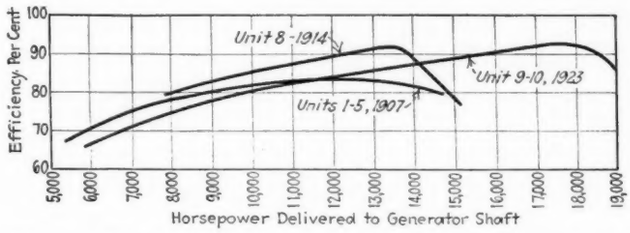


Fig. 9—Efficiency Curves of the Different Units Based on 54-ft. Head

the lower bearing bracket and bear against a machined braking surface on the lower side of the rotor. For braking, the jacks are operated by air pressure at 300 lb. and are designed to stop the unit in five minutes with turbine leakage sufficient to give a torque equivalent to 5 per cent of rated load. When used for lifting, the jacks will be operated by oil at a pressure of over 1,500 lb. per square inch.

The Determination of Starting Torque for Motor Drives

BY P. B. JAMESON*

In industrial work a large part of the electrical power supplied is used in alternating-current squirrel-cage induction motors. The ordinary three-phase squirrel-cage motor is ideal for power work where it meets the required conditions, because of its simple substantial construction. It is as near foolproof as any motor can be. Oftentimes there is some doubt about applying such a motor to a load where the starting torque is known to be large, and the question arises whether the motor will start the load. It is usually a fairly easy matter to determine the size of motor that will carry the load once it has been started, but the starting load generally determines the character of the motor that must be used.

The principal machines on which high starting effort is required are: Direct-driven wood-working machinery, power-transmission systems, air compressors, machinery equipped with flywheels, high-speed grinders, hoists and elevators.

The high starting effort of these loads is due either to high friction, heavy flywheels or high operating speeds. Any or all of these conditions may combine to cause a large starting torque. Before discussing a method for determining the starting torque of a load, it is well to understand the various terms used.

Starting torque is the turning effort required to start a given load. It is usually expressed in pounds at one foot radius, or foot-pounds. The starting torque of a motor is expressed in terms of the full-load torque as 150 per cent, or 1.5 times the full-load torque. Full-load torque is the turning effort required to keep full rated load of the motor in motion once it has been started and brought up to normal speed. The full-load torque of any motor can be determined from the formula,

$$Hp. = \frac{\textit{full-load torque} \times \textit{r.p.m.} \times 2\pi}{33{,}000} = \frac{\textit{Full-load torque} \times \textit{r.p.m.}}{5{,}250}$$

which may be transposed to read

$$\textit{Full-load torque} = \frac{Hp. \times 5{,}250}{\textit{r.p.m.}}$$

Thus, the full-load torque of a 3-hp., 1,800-r.p.m. motor could be,

$$\textit{Full-load torque} = \frac{3 \times 5{,}250}{1{,}800} = 8.75\ lb.$$

at one foot radius,

that is, with a 24-in. pulley on the armature shaft the motor when taking full load current would develop a pull of 8.75 lb. at the surface of the pulley.

Where there is any doubt as to the ability of a given motor to start the load, it should always be determined by actual tests. The method of making this determination is quite simple. The principal considerations which must be known about a load are: (1) The starting torque at one-foot radius; (2) the full-load speed of the machine or shaft to be started.

To determine the starting torque of a shaft, for example, first throw off the main belt. This leaves just the friction load of the belts and countershafts which must be started, together with the loose pulleys of the various machines. If several of the machines are on tight pulleys, they must be left so because that is part of the load which the motor will have to start. Suppose this shaft, which it is proposed to motorize, turns at 200 r.p.m. and it is known that when in operation a 3-hp. motor will have ample power to do the work, but there are several machines belted to this shaft, so that it is quite long, and there is some question whether a 3-hp. 1,800-r.p.m. three-phase squirrel-cage induction motor would start the shaft.

Having thrown off the main belts so that the conditions exist under which the motor must operate, fasten a pipe or bar of some kind of sufficient length in a vertical position to the main pulley or to any convenient pulley by which the shaft can be turned. Then near the outer end of the pipe at a definite distance from the center of the shaft apply force through a spring scale (see Fig. 1). This force, which is measured in pounds times the distance from the line of action of the force to the center of the shaft in feet, will give the foot-pounds starting torque, or pounds at one-foot radius.

If a spring scale is not available, fasten the pipe to the pulley in a horizontal position with the same length of pipe overhanging each side of the shaft. This is done so that a correction will not have to be made for the weight of the pipe or bar. Hang a known weight on one end of the bar near the pulley and slide it toward the outer end of the bar to a point where the shaft just begins to turn, repeating this to be sure of the location of the weight. Then the distance from the

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point where the weight applied from the center of the shaft in feet times the weight in pounds equals the foot-pounds, or pounds at one-foot radius (see Fig. 2).

In the case illustrated in Fig. 1 it is usually more convenient to choose a definite distance, and in Fig. 2, it is better to take a known weight and slide it toward the end of the bar to the right point. If, in the case represented by Fig. 2, it is not possible to get the overhang of the bar equal on each side of the shaft, a correction may be made as illustrated in Fig. 3. Referring to the figure, suppose the bar is 7 ft. long and weighs 10 lb. Then the unbalanced overhang is 5 ft., or $\frac{5}{7}$ of 10 lb., which equals $7\frac{1}{7}$ lb. The center of gravity of a uniform bar 5 ft. long would be $2\frac{1}{2}$ ft. from the end. Therefore, the weight of the overhanging bar will act at a distance of $3\frac{1}{2}$ ft. from the center of the shaft, as indicated at *G*. Thus the correction to be added to the

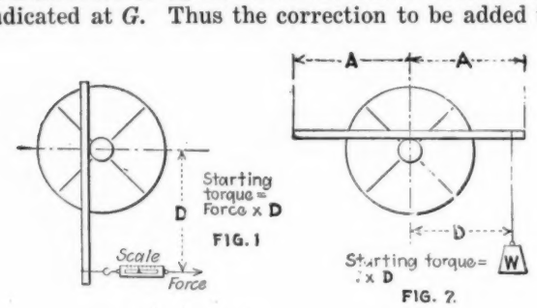


FIG. 1

FIG. 2

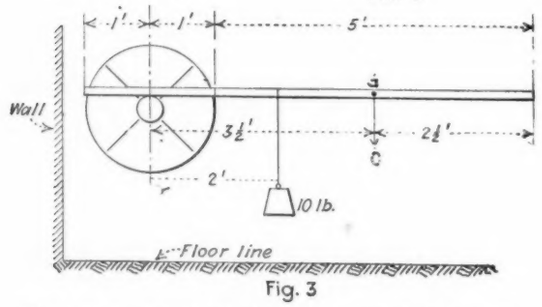


Fig. 3

Figs. 1 to 3—Methods of measuring torque

foot-pounds already found by multiplying the weight times the distance from the center of the shaft would be $7\frac{1}{7}$ lb. times $3\frac{1}{2}$ ft. equals 25 foot-pounds.

Now, to return to the shaft to be motorized, suppose there is attached to the main pulley a 7-ft. bar as illustrated by Fig. 3 and it is found that a weight of 10 lb. hung on the bar 2 ft. from the center of the shaft will just start it turning; then the starting torque is 10 lb. $\times$ 2 ft. = 20 ft.-lb., plus the correction of 25 ft.-lb., or 45 ft.-lb. in all. At 200 r.p.m. the starting horsepower required by the shaft would be:

$$Hp. = \frac{Torque \times r.p.m.}{5{,}250} = \frac{45 \times 200}{5250} = 1.71$$

The starting horsepower of a 3-hp. 1,800-r.p.m. motor may be determined as follows: Full-load torque equals 8.75 lb. at one-foot radius as previously determined. The average starting torque of an 1,800-r.p.m. squirrel-cage motor may be taken at about 1.5 times full-load torque. Therefore, starting torque equals 8.75 $\times$ 1.5 = 13.1 lb. at one-foot radius. Hence, the starting horsepower of the motor $= \frac{13.1 \times 1{,}800}{5{,}250} = 4.5$. It is evident from this calculation that the motor has ample starting torque to turn the shaft and its connected load.

Another and, to some perhaps, a clearer way to arrive at the same conclusion would be to bring in the pulley ratios. This, of course, was done in the previous calculations through the speed ratios. It has been assumed that it requires 45 lb. at one foot radius to turn the shaft in Fig. 3.

If the shaft turns 200 r.p.m. and the driven pulley is 2 ft. in diameter, then the motor pulley or driver would theoretically be 200 $\div$ 1,800 = $\frac{1}{9}$ of 24 in., or 2.66 in. in diameter, or 1.33 in. radius. The formula showed that a 3-hp. motor has a starting torque of about 13.1 lb. at one foot radius. At 1.33 in. radius the motor could develop a torque of 13.1 $\times$ 12 $\div$ 1.33 = 118 lb. As the load requires only 45-lb. pull at the surface of the driven pulley, it is evident that the motor has ample torque to start the load satisfactorily.

A pulley for a 3-hp. motor of 2.66 in. would, of course, be out of the question, so the size would have to be changed in practical operation to something like a 4-in. driver to a 36-in. driven. The ratio of 1 to 9 holds in any case, however.

A point that is interesting to note in this connection is that a general-service squirrel-cage motor of this size has a starting torque of about 40 per cent of full-load value at full-load current. So that if the voltage were reduced at starting to keep the starting torque in the neighborhood of full-load current as the motor picks up the load, it would have a starting horsepower of $\frac{0.4 \times 8.75 \times 1{,}800}{5{,}250} = 1.2$, which would not be quite sufficient, as 1.71 horsepower is required. Or to state it in terms of torque, it would have a starting effort of 0.4 $\times$ 8.75 = 3.5 lb. at one-foot radius, which is not enough, as 5 lb. is needed.

The accompanying table, which is reprinted from the "Condit Handbook on Starters," gives the approximate starting characteristics of alternating-current and direct-current motors. This table, while not absolutely accurate, is handy in helping to determine some of the starting problems that come up in motor installations. If real accuracy is demanded, the safest way is to obtain the starting characteristics of the motor in question from the manufacturer.

STARTING CHARACTERISTICS OF DIRECT-CURRENT AND ALTERNATING-CURRENT MOTORS
(Approximate)

Kind of Motor	Starting Torque in Per Cent of Full-Load Torque	Starting Current in Per Cent of Full-Load Current	Per Cent Full-Load Torque for Full-Load Current	Maximum Pulling Torque in Per Cent of Full-Load Torque
Small d.-c. comp. without starter, ¼-hp. and smaller	350	450	78	275
Small d.-c. shunt without starter, ¼-hp. and smaller	250	250	55	200
Large d.-c. comp. with starter, ½-hp. and larger	200	170	118	300
Large d.-c. shunt with starter, ½-hp. and larger	180	170	106	225
Small two- and three-phase squirrel-cage induction motors without starter, ½-hp. and smaller	215	475	45	225
Two- and three-phase squirrel-cage induction motors without starter, ½-hp. and larger	225	600	41	225
Two- and three-phase wound-rotor induction motors with resistance in rotor for starting, 5-hp. and up	150	150	100	225
Single-phase induction motor split phase start, up to ½ hp.	220	500	44	225
Single-phase induction motor with clutch and with hand or automatic start, for cutting resistance and reactance in and out of circuit up to 15 hp	140	250	56	150
Single-phase strongly compensated repulsion motor, up to 15 hp	225	500	50	275
Single-phase weakly compensated repulsion motor, up to 1 hp	360	270	133	300
Single-phase, repulsion-start induction motors $\frac{1}{10}$ to ¼ hp. inclusive	450	260	175	225
Single-phase repulsion-start induction motors, ½ hp. and larger	335	270	125	175
Single-phase repulsion-start induction motor 7½ hp. and larger, with resistance starter	100	170	60	175
Single-phase repulsion-start induction motor, 7½ hp. and larger, with compensator starter	100	100	100	175

Proportioning Water-Works Surface Condensers

ASSEMBLY of formulas and curves that may be used to advantage in the proportioning of water-works condenser with typical example to demonstrate their use.

BY W. R. FREEMAN

IN STEAM-DRIVEN power plants where the pumping of water rather than the generation of electricity is the chief function, relatively large quantities of water are pumped as compared with the steam to be condensed from the driving unit. All this water is available for condensing purposes, the ratio of circulating water to condensed steam being high, and consequently the temperature rise of the water low.

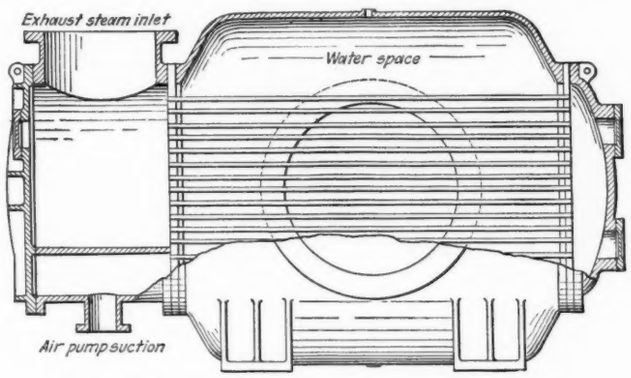


Fig. 1—Section through water-works condenser

Owing to the high water ratio and the large quantities available for circulating purposes, the standard type of surface condenser in which the water is passed through the tubes would not be suitable, as high friction head would be imposed, and since this friction head would apply to the entire quantity of water, there would be an economic waste. This can be avoided by providing a large condenser shell and circulating the water around the tubes at comparatively low velocity and passing the steam through the tubes.

This design, as distinguished from the more usual type, is called a water-works condenser and is shown in section in Fig. 1. The condenser shell is made considerably larger than the tube sheet to provide additional areas and decrease the water velocity and the friction head, which must be kept at a minimum, for in many cases the friction head may become an appreciable percentage of the total pumping head.

To compute the surface required for condensing the steam, the following formula may be used:

$$S = \frac{WL}{U\left(T_s - \frac{T_1 + T_2}{2}\right)} \qquad (1)$$

where

S = Surface in square feet;
W = Pounds of steam to be condensed per hour;
L = Latent heat per pound of steam rejected to the condenser, usually taken as 1,000;
T_s = Temperature of the steam corresponding to the vacuum;
T_1 = Temperature of the circulating water at the inlet;
T_2 = Temperature of the circulating water at the outlet;
U = Heat transfer in B.t.u. per hour, per square foot, per degree difference between temperature of steam and mean temperature of water, U being taken usually at not to exceed 300, due to the low water velocity and the low water temperature usually available.

Where GPM equals gallons per minute of circulating water available,

$$T_2 = T_1 + \frac{2W}{GPM} \qquad (2)$$

Since the steam passes through the tubes, a long travel must be avoided or there will be excessive steam friction and vacuum drop between the exhaust-steam inlet and the air-pump suction. For this reason 1-in. tubes are selected and they should never exceed 10 ft. in length. The steam box is arranged for two passes of the steam through the tubes. During its flow through the condenser the volume is reduced by condensation, so that the second, or return, pass need not have as much area as the first pass. For the foregoing reasons the

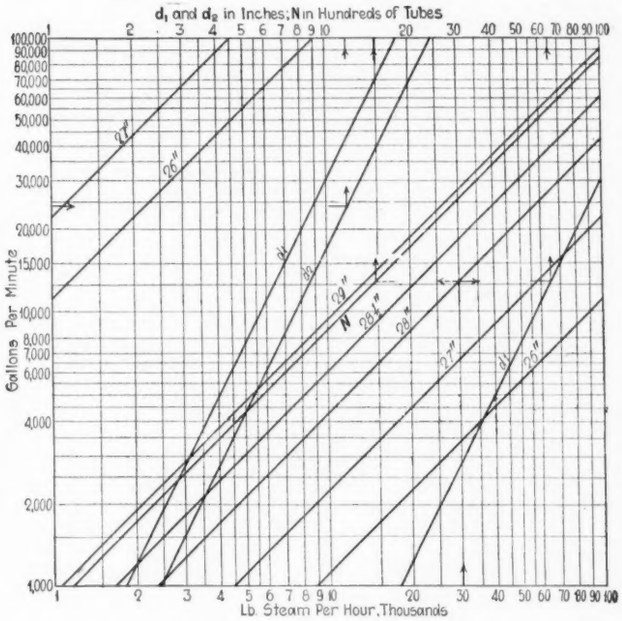


Fig. 2—Chart giving number of tubes and diameter of tube sheet and shell

total areas of the first and second passes are usually in a ratio not to exceed 6 to 1.

In the first pass the arrangement of the tubes should be such that the steam velocity at the entrance of the tubes will not exceed the values given in the table.

With these limitations the number of 1-in. tubes in the first pass will be

$$n = KW \qquad (3a)$$

where K is taken from the table and W represents the pounds of steam per hour to be condensed.

Since the tubes in the two passes should be in the

ratio of 6 to 1, then the number of tubes in the second pass should be $\frac{n}{6}$ and the total number of tubes

$$N = \frac{7n}{6} \qquad (3b)$$

In the tube sheet the tube spacing should not exceed 65 tubes per square foot of tube-sheet area measured

STEAM VELOCITIES AT ENTRANCE TO TUBES IN FIRST PASS

Vacuum, In.	Velocity, Ft. per Sec.	K
26	600	0.0110
27	400	0.0220
28	300	0.0425
28.5	275	0.0600
29	250	0.0900

at the inside of the steam box, and the tube-sheet diameter in inches exclusive of flanges (that is, steam-box diameter) may be obtained from the formula,

$$d_1 = 1.815\sqrt{n} \qquad (4)$$

As stated, the shell diameter must be increased to permit lower water velocities and to keep the friction head at a reasonable figure. Practice has limited the friction head through the condenser to a value not to exceed 2 ft. This low friction head may be obtained by enlargement of the shell to provide an annular space between the shell and the tubes. The increase in diameter in inches of the tube plate may be obtained from the formula,

$$d_2 = 0.0766\sqrt{GPM} \qquad (5)$$

Then the inside diameter of the shell in inches will be $D = d_1 + d_2$.

Equations 1, 3, 4 and 5 have been charted in Fig. 2. This chart is to be used after the surface has been obtained from formula (1). With the surface, vacuum and *GPM* known, d_1, d_2 and N are obtained as indicated by the arrows.

Finding Number of Tubes and Diameter of Tube Sheet and Shell

Enter the chart at the bottom at the quantity of steam to be condensed, proceed vertically to the vacuum line, thence horizonally to d_1, either to the right or to the left, and vertically to the diameter of the tube sheet d_1. Also, from the intersection of the steam quantity and vacuum line, proceed horizontally to the "N" line and then vertically to obtain the number of tubes. To obtain the shell diameter, enter the sheet at the left side at the quantity of circulating water, proceed horizontally to d_2 and then vertically to obtain the inches to be added to the tube sheet. Adding d_1 and d_2 will give the shell diameter D.

The required length of tubes will be,

$$L = \frac{S}{N} \times 3.82 \qquad (6)$$

The nearest even length in half-feet may be used where the results are in odd inches, but in no case should the length exceed 10 ft. or an excessive vacuum drop may be expected.

After final selection of the tube length the total number of tubes should be rechecked, using the value 0.2618 sq.ft. of surface per foot of length of 1-in. O.D. tube.

The length of the steam box should not be less than 0.8 of the diameter of the exhaust-steam opening.

Tubes should be rolled into the tube sheet at one end, preferably the inlet end, to prevent their turning by the action of the water, and should be pitched to provide a natural drainage and flow with the steam, thereby preventing accumulation of condensate in the tubes.

Water openings into and out of the shell should be of such size that the velocity of the water will not exceed 8 ft. per sec., or

$$Diam. = 0.23\sqrt{GPM} \qquad (7)$$

and may be located in the shell to suit conditions, except that provision should be made to prevent short-circuiting of the water.

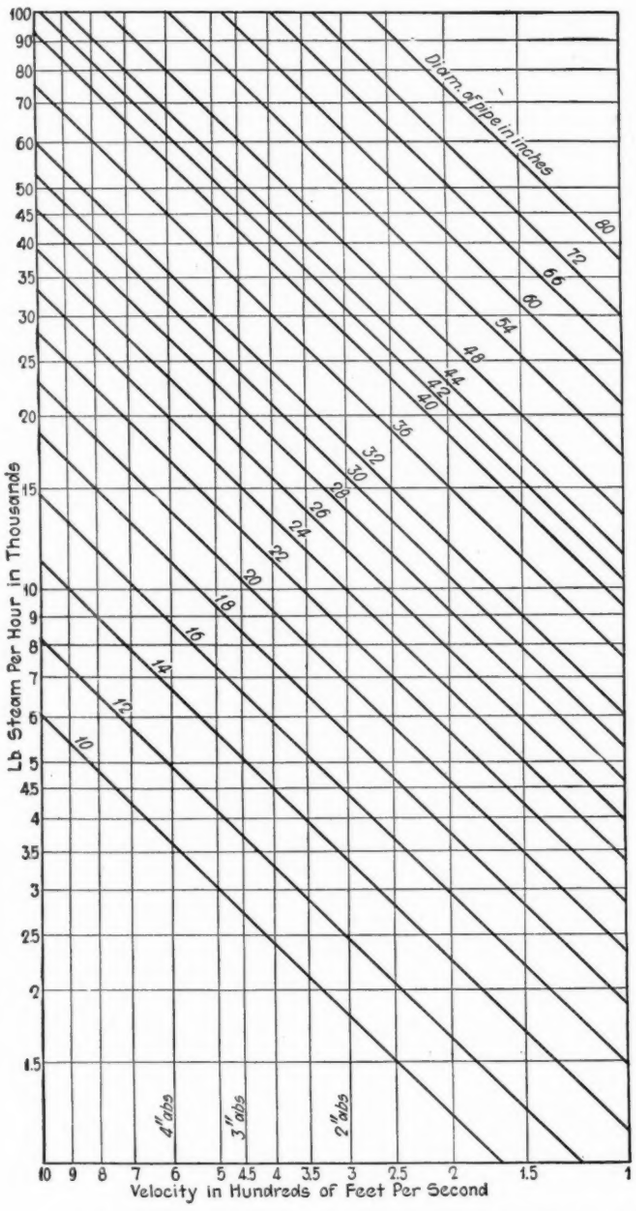


Fig. 3—Diameters of exhaust-steam opening

In this chart the steam flow in the exhaust lines is based on a condenser pressure of 2 in. abs. For other pressures multiply by the following factors:

Vacuum, in. abs.	1	1.5	2	2.5	3	4	15	30
Factor	0.516	0.761	1.0	1.233	1.463	1.919	6.63	12.67

Basis of foregoing dry steam. Increase the capacity by the percentage of wetness.

A typical example will illustrate the application of the foregoing. A water-works condenser is desired with capacity to condense 30,000 lb. of steam per hour and produce a vacuum of 28 in., referred to a 30-in. barometer (102 deg. F.). The pumps are handling 34½

million gallons per day of 24 hours, or approximately 24,000 gal. per min., at a temperature of 60 deg. F. The water ratio is

$$\frac{24{,}000 \times 8\frac{1}{3}}{\frac{30{,}000}{60}} = \frac{24{,}000 \times 500}{30{,}000} = 400$$

Assume the latent heat of the steam rejected to the condenser as 950 B.t.u. per lb., then the temperature rise is $\frac{950}{400} = 2.37$ deg. and the outgoing water temperature T_2 is $60 + 2.37 = 62.37$ deg. F. From formula (1) the required surface for the duty specified is

$$S = \frac{30{,}000 \times 950}{300\left(102 - \frac{60 + 62.37}{2}\right)} = 2{,}327 \; sq.ft.$$

From formula (3a) the number of 1-in. tubes in the first pass is,

$$n = 0.0425 \times 30{,}000 = 1{,}275$$

and the total number of tubes,

$$N = \frac{7}{6} \times 1{,}275 = 1{,}487$$

The diameter of the steam box or effective tube plate will be from formula (4),

$$d_1 = 1.815\sqrt{1{,}275} = 64.8, \; say \; 65 \; in.$$

The increase in diameter of the tube sheet is

$$d_2 = 0.0766\sqrt{2{,}400} = 11.87, \; say \; 12 \; in.$$

Then the diameter of the shell is

$$D = 65 + 12 = 77 \; in.$$

From formula (6) the tube length is,

$$L = \frac{2{,}327}{1{,}487} \times 3.82 = 5.98, \; or \; 6 \; ft.$$

The correct number of tubes is then $\frac{2{,}327}{6 \times 0.2618} = 1{,}481$. The water opening in the condenser shell, from formula (7) should not be less than $0.23\sqrt{24{,}000} = 35.65$, or 36 in.

Usually the length of the steam box is determined by the size of the exhaust opening of the turbine or engine, but should not be less than 0.8 the diameter necessary to pass 30,000 lb. per hour at a velocity of 300 ft. per sec. for a 28-in. vacuum. From Fig. 3 the diameter is 42 in. and $0.8 \times 42 = 33.6$, or say 34 in.

This completes the proportioning of the condenser from formulas and data that have been proved in practice and may be used to advantage in the design of a water-works condenser.

It is a matter of great importance to users of steam turbines that the steam be clean. In the first stages, where the steam is dry, only a small amount of foreign matter will be deposited, as this will be blown through in the form of a dry dust which may, however, cause erosion of the blading. At later stages, where the steam becomes moist, there is an increasing tendency for a paste to be formed which may set hard when the turbine is at rest. This may partly or even completely fill up the spaces between the blades and, in addition, collect at the bottom of the casing. If the quantity of deposit present be large enough, an additional bending stress will be imposed on the blades when the turbine is started. Again, the increase of the weight, owing to the choking of the passages, leads to an increase of the centrifugal force on the blades, which may raise the total stress to an undesirable figure.—*From Technical Report of British Engine, Boiler & Electrical Insurance Co., Ltd.*

Four Years' Development in Ice-Making and Refrigerating Plants*

By Halbert P. Hill

Mechanical refrigeration is a comparatively recent commercial development. In about 1755 the first experiments were made with a view of producing refrigeration by mechanical means, at which time temperature-pressure relations of various refrigerating mediums were observed. During the period from 1755 to 1823 many experimental machines were constructed, but the real foundation for the development of both compression and absorption machines was made in 1823 with the discovery that certain liquids, on being compressed to a higher temperature, would liquefy when cooled. Farraday developed a theory of the change of state of vapors in 1823 and thus opened up the way to the production of refrigeration.

In 1834 one of the first compression machines was developed by Jacob Perkins, famous for his invention of the unaflow engine and the high-pressure boiler. John Gorrie, of Florida, developed a cold-air machine in 1850. In 1855 Ferdinand Carre, of France, developed an absorption machine, although this was not offered commercially until 1865, at which time the first transparent ice was made from distilled water, in the United States. From 1873 to 1890 a large number of inventors, including Linde, of Germany, and David Boyle, of the United States, produced commercially successful machines.

Development Has Been Slow

While during the last 20 years the art of mechanical refrigeration and ice making has made considerable progress, unfortunately its development has been slow compared to that of the steam turbine, oil engine, automobiles, etc., from the fact that in most northern localities there was an abundance of natural ice, the price of which was low and the cost of transportation reasonable. The field was not sufficiently attractive to encourage research work, and the development was slow, but with several off seasons and the corresponding shortage of natural ice, the demand for artificial ice increased. Since a certain amount of distilled water had to be made to make sufficient ice, very little attention was paid to the economy of the boiler and less to the economy of the engine, as in reality the latter acted as a reducing valve since all of its exhaust was used in making distilled water.

However, with the development of raw-water ice systems the question of power became serious. More attention is given to prime movers, and efficiencies of 70 to 75 per cent are easily obtainable in the boiler room.

The ammonia compressor has passed through a period of evolution and the old slow-speed machine is being rapidly supplanted by the more modern high-speed machines, and the two-stage or compound compressor is rapidly coming into general use.

Development of the high-speed compressor has brought about improvements in electrical apparatus. In order to be able to operate at varying conditions of pressures and capacities, the clearance pocket compres-

*Abstract of paper read before the National Association of Stationary Engineers, Grand Rapids, Mich., Sept. 11, 1924.

sor has been developed, and this permits the driving of the compressor by a single-speed motor such as the synchronous type.

Early synchronous motors were started by means of an induction motor which brought them up to speed, and then thrown across the line and pulled into step, after which the fields were energized and the motor operated at synchronous speed. These synchronous motors had very little starting torque, and this called for the development of a self-starting synchronous motor, the machines now being designed with a 35 per cent starting torque and a pull-in torque of from 18 to 30 per cent

Advantages of Compound Compressor

The compound compressor has come into use and has many advantages. It lends itself more easily to electric drive, better volumetric efficiency is obtained, and by intercooling between stages with water, a saving in power is made possible, which in turn means a high over-all efficiency. While compound compressors have been used in many refrigerating plants, it is only recently that it has been installed in ice-making plants. It has proved very economical in the South, where high head pressures have to be contended with. The compression being divided into two steps, the small pressure difference in any of the cylinders permits a high volumetric efficiency to be attained. The mechanical efficiency of the two-cylinder machine compares favorably with, and in many instances is higher than, the efficiency of the single-cylinder compressor of the same capacity. The fluctuations caused by flywheel effect is more easily compensated for.

With the development of the feather, ribbon and ring-plate valve higher speeds have been made possible, and efficiencies have been increased considerably owing to the larger valve area and the light weight of the valve as compared with the old-type valves which in some cases weighed as much as 25 lb. Increased valve area has done more to raise the efficiency of the compressors than anything else.

The development of the oil engine as a prime mover in the ice plant has been slow, for the reason that an ice plant must operate continuously. Where an oil engine is installed, it is advisable to have duplicate units, and when this is not possible the engine should be of ample size. As a matter of fact it is good practice to install an engine 25 per cent larger than load requirements.

Ammonia Condenser Design

The ammonia condenser, one of the most abused pieces of apparatus in the ice-making or refrigerating plant, has been designed in every conceivable shape and type. The atmospheric condenser is the type most generally adopted, while the double-pipe condenser has met with favor in many instances on account of its ability to be placed on the wall, in the engine room, or in any convenient locality.

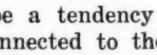
The shell-and-tube condenser is one of the latest developments and has many advantages, from the viewpoints of economy and ease of operation. When properly installed, it has a higher efficiency than other types. Being built both horizontal and vertical, it is well to appreciate the advantages of each of these styles. The horizontal design should be used only where the water has to be pumped through the condenser, or where there is no available space, necessitating them to be installed one above the other. They have the disadvantage that the heads must be removed to clean the tubes. While there are many successful installations of this type, yet the vertical shell-and-tube condenser is much more efficient, and where there is space to install it, it makes the ideal machine. Water flows down the tubes at high velocity, a thin film covering the inside surface of the tube, and the discharge is into a pan or sewer connection at the bottom of the condenser. The condenser can be cleaned in operation, and data from numerous installations that have been in operation for more than six years, indicate that it requires few or no repairs and that the life is exceptionally long and the efficiency extremely high. Recent tests made on this type of condenser showed a heat transfer of 231 B.t.u. per sq.ft. per hour per deg. of temperature difference, and do doubt a higher transfer could have been obtained save that the test was limited to the amount of water that could pass under the guide ring. In another test a heat transfer of 315 B.t.u. was obtained with 4½ sq.ft. of cooling surface per ton of refrigeration.

Progress in Raw-Water Ice Manufacturing

There is probably no process that has been so slow in development as raw-water ice manufacturing. The quality of the water has a material bearing on the clearness of the ice. With good water, the ordinary drop-pipe low-pressure air system can produce clear marketable ice. The main disadvantages are the sucking of the cores and the removal of the air tube, both of which are labor problems. With high-pressure systems the air is kept on until a can is completely frozen; for this reason it makes a better ice with inferior water. Most of the high-pressure systems are alike in principle, and the main difficulty lies in drying the air. With the well-designed dehumidifying system a large part of this trouble is eliminated.

Few freezing tanks have sufficient agitation, and very few are properly piped, since one must have sufficient cooling surface for the brine coolers. It has been found that 30 sq.ft. of cooler surface per ton of ice is none too much, and there are many plants installed having as low as 18 ft. Where coils are employed, they should be connected up so that the freezing tank can be operated flooded. This cannot be done unless means are provided to prevent slopping over and freezing back to the machine.

The introduction of accumulators set in the freezing tanks, in a manner accomplished this purpose, but the modern method is to place the accumulator at a convenient point below the freezing tank. It is piped up with the cooling coils, and an ammonia pump is installed, so that any liquid that slops over into the accumulator can be taken off by the pump and delivered to the high-pressure side of the system.



There seems to be a tendency to install auxiliary generators direct connected to the shaft of the main unit. This has the advantage of reducing the number of elements to be governed. There is undoubtedly less trouble with the large and rugged governors than there is with the small ones. In case of interruption of the power demands on the main unit, this arrangement would always provide a small load to the main turbine. It has, however, the disadvantage of requiring more space.—*Prime Movers Report, N.E.L.A.*

Steam Turbines*

BY FRANCIS HODGKINSON

The problem of designing the steam path has changed from one of securing reasonable efficiency at low rotative speeds to that of securing sufficiently large areas of the last rows of blades. Both high- and low-pressure blading and a novel means intended to free the turbine from part of the water in the low-pressure region are discussed.

WHILE intended to cover the tendencies of American practice in steam turbines, this is written from the viewpoint of a representative of one of the principal manufacturers of the United States who began building under a license from C. A. Parsons & Co. in 1895. It is therefore written with less intimate knowledge of the practice of other American builders. Because of restrictions of space the subject matter is limited to large turbines as used by public-utility companies, and further limited to the turbine proper. Therefore, detailed discussion of lubricating, regulating and such auxiliary apparatus is excluded.

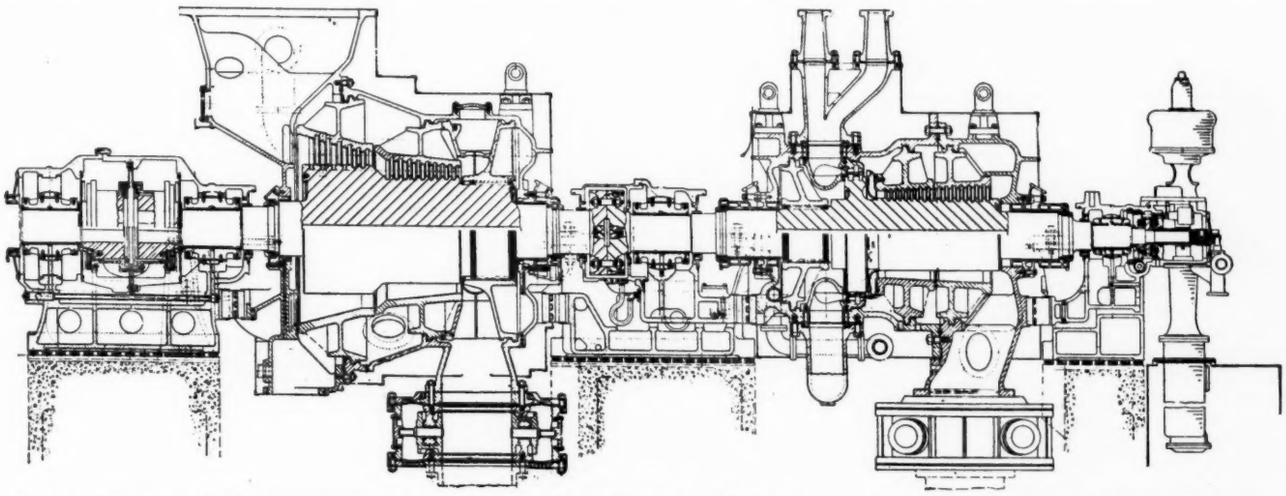


Fig. 1—Thrust of high-pressure rotor balances that of intermediate pressure, making dummy pistons unnecessary

COMPOUND TURBINES LIKELY FOR HIGH PRESSURES AND LARGE CAPACITIES

The history of turbine progress, at least with the C. A. Parsons Co. in Europe and the Westinghouse company in America, has been that with increases of capacity builders have turned to compounded turbines, either tandem or cross-compound, and as the art progressed, they have subsequently produced the same capacities in single-cylinder units.

It would seem, however, at least from the viewpoint of today, that turbines of 50,000-kw. maximum continuous capacity and greater and for turbines of almost any large size where steam pressures higher than 400 lb. are employed, the compound principle in one form or another will be adhered to. The General Electric Co. has, however, produced eminently successful machines of 50,000-kw. capacity at 1,200 r.p.m. as single complete expansion machines for the Detroit Edison Co. and is installing another for the Brooklyn Edison company.

Impulse turbine elements used to be regarded as highly advantageous in lending themselves to fractional admission of steam—that is, the nozzles occupying a fraction of the circumference—and were appropriate to low rotative speed turbines. This involved at least two serious losses, one the windage of idle blades, the other known as displacement loss, which is the energy required to sweep out the inert steam from the blade passages when they come within the active arc of steam admission.

Today, in all impulse-turbine design, nozzles are made to occupy as much as possible of the whole circumference. This displacement loss for each nozzle group is expressed as follows:

$$W_d = \frac{8.5\ hb\ V_w}{S_v}$$

where

W_d = Displacement loss in pounds of steam per hour;
h = Mean blade height in inches;
b = Blade width in inches;
V_w = Mean blade velocity in feet per second;
S_v = Specific volume corresponding to steam condition in impulse wheel chamber.

This explains why individual or group control of nozzles does not produce the gain in efficiency sometimes expected. The greater the arc occupied by the nozzles the smaller the blades will be, resulting in both lower displacement and windage losses.

In any turbine blading the larger the blade passage the greater the efficiency because of the lesser ratio of perimeter to area, and hence, less friction loss. Because of higher steam velocities and greater energy conversion in impulse elements any departures from ideal blade form are more marked in results as compared with reaction elements.

*Abstract of a paper presented at the World Power Conference in London, June 30-July 12, 1924.

The opinion is generally held that nozzle efficiency falls off more rapidly after passing the critical velocity. Some experimenters have held the opinion that maximum efficiency occurs at that velocity and there is a good deal of conflicting data on the subject, much of which is based on experiments with nozzles discharging against a target.

In an actual turbine, however, blade losses may be totally different from experimental data of this kind because of the effect of the moving edges, displacement, entrainment, etc. Many factors may obscure results. For instance small turbines with wide blades, having small arcs of admission, might show superior results from such experimental tests, while in practice they would be less efficient because of high displacement loss.

It would seem evident that the highest blade efficiency is secured with lowest velocity, and that impulse-blade efficiency is affected by the ratio $\frac{T}{R}$, which is the ratio of the steam path thickness between blades to the radius of curvature of the concave surface of the blade.

Thus, $\frac{T}{R} = \frac{\text{width of passage between blades.}}{\text{radius of concave blade curvature.}}$

Edge thickness loss is greatest with low values of $\frac{T}{R}$.

Loss due to friction on the blade, compression effects of the steam due to reversal of direction, stream interference and "straight blow-through," all increase with high values of $\frac{T}{R}$. Consequently, it is evident that as the edge thickness loss decreases with increase of this ratio, a point of minimum loss occurs at some relatively low ratio value for each steam velocity and blade width.

In practice, curves have been determined by the analysis of results from actual impulse wheels. The edge loss remains constant for any value of $\frac{T}{R}$. The velocity losses increase with the higher velocities; hence this ratio should have lower values with higher steam velocities. By increasing the width of blades in an axial direction further reduction of loss may be secured with lower ratio values because of the reduced edge loss, and with fewer edges passing in front of the nozzle less disturbance of the stream lines occurs.

The values of $\frac{T}{R}$ employed in Westinghouse turbine practice for two-row impulse elements vary from three-tenths in the first moving row to six-tenths in the second moving row for steam nozzle velocities of approximately 2,000 ft. per sec. Practical designing and the employing of desirably low ratio values lead to wide blades and have resulted in Westinghouse impulse blades being wider than those of other manufacturers.

The results obtained with blading designed with the foregoing points in view seem to justify the conclusions given. A notable point is that with this form of blade an erosive cupping action so often seen within the blade passages of narrow impulse blades has not been observed even with many years' operation with wet steam and steam velocities exceeding 3,000 ft. per sec. It is obvious that a given degree of erosion is far less detrimental to the efficiency of a wide blade than a narrow one.

The experience of the General Electric Co. has been a progress from a turbine comprising two stages each with three or four velocity extractions to a greater number of stages with two velocity extractions, known as Curtis turbines, finally reaching their highly successful turbine comprising one two-row velocity stage at the high-pressure end and followed by single velocity stages of the order of Rateau elements for smaller capacities at given r.p.m. For the larger machine single-velocity stage elements only are used, of which a large number have been installed.

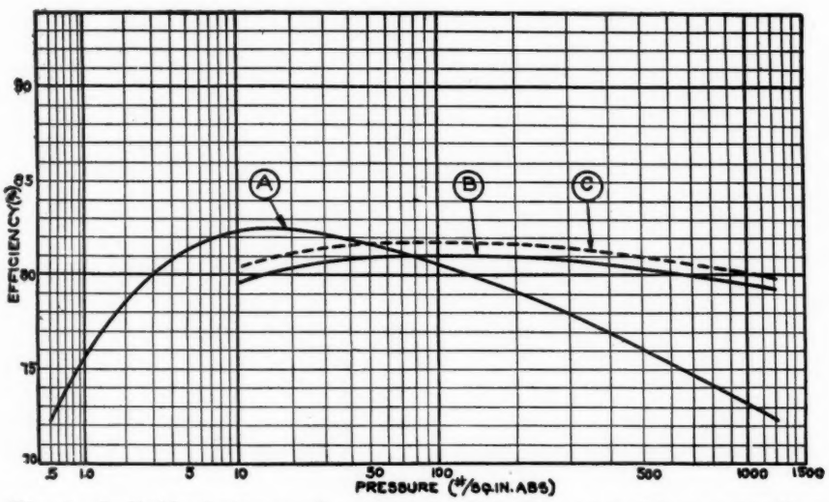


Fig. 2—Individual stage and combined efficiencies of turbines expanding from 1,200 lb. to 29 in. vacuum

At *A* is shown expected efficiencies for individual stages of pressure on horizontal scale, 700 deg. *F.* steam, no reheating. Efficiencies of a turbine from these pressures to 29 in. vacuum as expected, at *B*. Curve *C* same as *B* except that entire steam is extracted at about one-third the B.t.u. drop and reheated to 700 deg.

Their latest 50,000-kw., 80 per cent power-factor, 1,200-r.p.m. turbine is designed to operate with 265-lb. pressure, 200-deg. superheat and 1-in. abs. back pressure. At 40,000 kw. the guaranteed steam consumption is 10.05 lb. per kw.-hr., which is equivalent to an efficiency ratio of 77.67 per cent including all losses except those on account of extraction for feed-water heating. Provision is made, however, for bleeding approximately 75,000 lb. per hour from the tenth stage and 45,000 lb. from the fifteenth stage. The overload valve bypasses into the sixth stage.

The turbine comprises twenty-one single-row impulse stages, the mean diameters of elements increasing from 60 in. for the first wheel to 136½ in. for the last wheel. The last blades are 30½ in. long. The progressive increase of diameter from stage to stage without abrupt changes in diameter provides a continuous steam path minimizing eddies and conserves residual velocities to the maximum degree. The small diameter of high-pressure wheels permits the nozzles to occupy the whole circumference.

The diaphragms from the first to the sixteenth stage are made of steel plate with removable nozzles. The last five diaphragms are of cast steel with alloy steel partitions cast therein.

The experience of the Westinghouse company began with reaction turbines of the Parsons type of that day. They later substituted an impulse element for the high-pressure portion of the reaction turbine, employing double-flow low-pressure elements for the larger capacities. Later they developed cross-compound turbines for the largest capacities and substituted the multiple exhaust blading for the double-flow low-pressure element.

The Allis-Chalmers company has not departed from the reaction type since beginning the manufacture of turbines.

Modern Westinghouse 35,000-kw., 1,800-r.p.m. turbines and others of similar design of different capacity are under construction and in course of installation. One of the features of design in which they differ from older practice is the employment of multiple exhaust blading. These blades are of warped surface and of tapering section so that the maximum stresses in the blade due to centrifugal force do not exceed 20,000 lb. The average stress in the drum is 18,000 lb. Both are calculated on a basis of 20 per cent overspeed.

The steam expands through the nozzles from initial pressure to approximately 120 lb. abs., the mean speed of the impulse element being 463 ft. per sec. The impulse blades are wide for the reasons stated in the foregoing, their widths being 2½, 2 and 1½ in., respectively, for the three rows of blades.

Erosion of turbine blading shows plainly that the moisture in a turbine occupies the outer zone of the steam path, so these later machines are arranged with the adjacent edges of the blade rings brought close together so that the velocity of the steam going to the heater at full load approximates 150 ft. per sec. It is believed that some proportion of the moisture will be carried away when the turbine is operating with full flow to the feed heaters. The effectiveness of this will be definitely determined by carefully conducted trials.

Reducing Moisture in Low Pressure Region

A three-cylinder unit being built by the Westinghouse company contains the high-pressure and intermediate arranged as a tandem-compound, 1,800 r.p.m. machine, is shown in detail in Fig. 1. They are arranged with steam flow opposed to each other so the one may neutralize the thrust of the other, eliminating the necessity of large dummies or balance pistons. At full load, the high pressure expands to 132 lb. abs., at which pressure it is reheated and returned to the intermediate turbine element where, at full load, it is expanded to 16½ lb. abs., passing to the low-pressure element driving its separate generator. Expectations of turbine performance of these machines have been based on a 10-lb. pressure drop through the reheater and its piping. It is important that every precaution be taken to keep this at a minimum. Should this be as high as 25 lb., practically all gain in economy from reheating would be lost.

The problem of the design of turbine blading, however, is not so much one to provide strength to resist centrifugal forces, but rather of sufficient knowledge of the subject to be able to eliminate the possibility of nodal vibrations. All turbine blades are subjected to the following conditions:

Bending moments because of the force causing rotation.

Bending moments incidental to the contraction and expansion of shrouding or lashing wires because of the blading heating and cooling at a different rate to the main blade carrying element.

Vibration stresses incidental to errors of balance and of frequency corresponding to the r.p.m.

Probable vibratory stresses produced by high period vibrations in the steam current.

Torsional vibrations incident to the character of the load, with alternating-current generators incidental to unbalanced phases, and a multiple of the r.p.m., are particularly manifest in geared turbines for marine propulsion and in the case of centrifugal pump drives.

To the foregoing are added the stresses incidental to centrifugal force in the case of the moving blades, and for this reason, breakage of moving blades is more common than stationary ones. This fact suggested the later Westinghouse practice of making moving reaction blades of greater width of section than their complementary stationary blades.

The desiderata of turbine blading materials are: (1) High fatigue limit; (2) a fine and uniform microstructure, not subject to internal changes or growth under long periods at elevated temperatures; (3) maintenance of physical strength and modulus at elevated temperatures; (4) non-corrosiveness.

Consideration must be given to hot-shortness of material if any soldering or brazing is to be performed. If hot-shortness occurs at or below soldering temperature, fracture will result if any stresses exist during cooling of the parts.

Some slight reduction of bearing losses without affecting reliability may be secured by raising the unit pressures on bearings to at least three times present practice, which, however, may not necessarily effect a reduction of the total length of the turbine unit.

Some gyroscopic stabilizer rotors, for example, have been operated successfully with bearing pressures many times higher than customary with turbine practice. In one instance a stabilizer rotor bearing 18½x28 in. operated successfully with a maximum pressure of 900 lb. per sq.in. of projected area and an average pressure of 720 lb. during one direction of precession. It, however, had the advantage of the journal moving over to the opposite side of the bearing at each precession.

If present turbine practice of about 150 lb. per sq.in. is increased materially, a means of establishing the oil film before the rotor is revolved is a necessity in order to avoid injury during the first fraction of a revolution. This has been readily accomplished in some cases by the temporary application of high-pressure oil in some appropriately disposed grooves in the bearing, establishing the oil film before motion begins.

It is not believed that the designing of turbines for pressures up to 1,500 lb. will present any insurmountable difficulties, although no practical experience has been had of the erosive action of steam at such pressures. Low steam velocities will, however, be employed generally. It is thought that total temperatures of 750 deg. F. will not be exceeded.

Concerning turbine efficiencies with high-pressure steam the curves of Fig. 2 have been prepared. Curve *A* shows the approximate stage efficiencies to be expected of the individual turbine stages of a turbine of modern design with 1,200 lb. sq.in. initial steam pressure exhausting against 29-in. vacuum. Curve *B* shows the average efficiencies of turbines of modern design from the pressures in the abscissa to 29-in. vacuum. Curve *C* shows the effect of reheating to the initial temperature at one-third of the heat drop.

How Much Work Can Be Obtained from a Pound of Fuel?

Rankine Cycle Obsolescent as Efficiency Basis—New Standard Needed—Proposed Method Based on Step-by-Step Application of "Second Law"—Rankine Comparison Misleading in Case of Regenerative Cycles and of Steam-Mercury Plants

BY LIONEL S. MARKS
Professor of Mechanical Engineering, Harvard University

HOW much work is it conceivably possible to obtain from the combustion of one pound of fuel? If the fuel is burned under a boiler and the steam generated is used in a steam turbine, we know that over-all efficiencies of 20 to 22 per cent can be obtained. From the mercury-vapor turbine Mr. Emmet (*Mechanical Engineering*, May, 1924) estimates 28.4 per cent under unfavorable conditions, and Kearton (Proc. Inst. Mech. Eng., 1923) estimates 32.3 per cent. For Otto-cycle engines, such as airplane engines, brake thermal efficiencies of 28 per cent and indicated thermal efficiencies of 33 per cent are obtained. In four-stroke-cycle Diesel engines, brake thermal efficiencies of over 35 per cent are obtained, with corresponding indicated thermal efficiencies of about 46 per cent. The highest thermal efficiencies ever recorded are for the Still engine, which consists of a Diesel engine with exhaust gases generating steam which acts on the under side of the Diesel piston; the record shows a brake thermal efficiency of about 40 per cent and an indicated thermal efficiency of about 50 per cent.

METHOD OF DETERMINING POSSIBLE EFFICIENCY

How near are these efficiencies to the maximum that theory shows to be possible? To answer that question, it is necessary to recall that the possible efficiency in converting heat into work depends upon two temperatures; these are (1) the temperature T_1 at which the heat is available, and (2) the temperature T_2 of the cooling substance, usually water or air, to which the heat is rejected. In a steam engine or turbine without superheat these correspond to the steam temperature and the condenser temperature, respectively.

The efficiency of conversion of heat into work is then given by the efficiency of a Carnot cycle, $\frac{T_1 - T_2}{T_1}$, where the temperatures are both expressed in the absolute scale by adding 460 to the usual Fahrenheit temperatures. If t_1 and t_2 are the same temperatures on the usual Fahrenheit scale, efficiency can be reckoned as equal to $\frac{t_1 - t_2}{T_1}$. The work done is the product of the efficiency by the heat supplied Q, or

$$\text{Work} = \frac{t_1 - t_2}{T_1} \; Q \text{ B.t.u.}$$

$$= \frac{t_1 - t_2}{T_1} \; Q \times 778 \text{ ft.-lb.}$$

Suppose a pound of fuel is burned in a Dutch-oven furnace, with no heat loss to the outside or to any heating surface. The temperature reached by combustion will depend on the amount of excess air present. For example, if the fuel is carbon, with a heating value of 14,500 B.t.u. per lb., the minimum air requirement is 11.52 lb., and if there is present 50 per cent excess of air, or $1.5 \times 11.52 = 17.28$ lb. of air, there will be 18.28 lb. of chimney gases, consisting of 3.67 lb. CO_2, 1.33 lb. O_2 and 13.28 lb. N_2. To find the heat required to raise these products of combustion one degree, multiply each by its specific heat at constant pressure: The heat required is $3.67 \times 0.21 + 1.33 \times 0.217 + 13.28 \times 0.247$

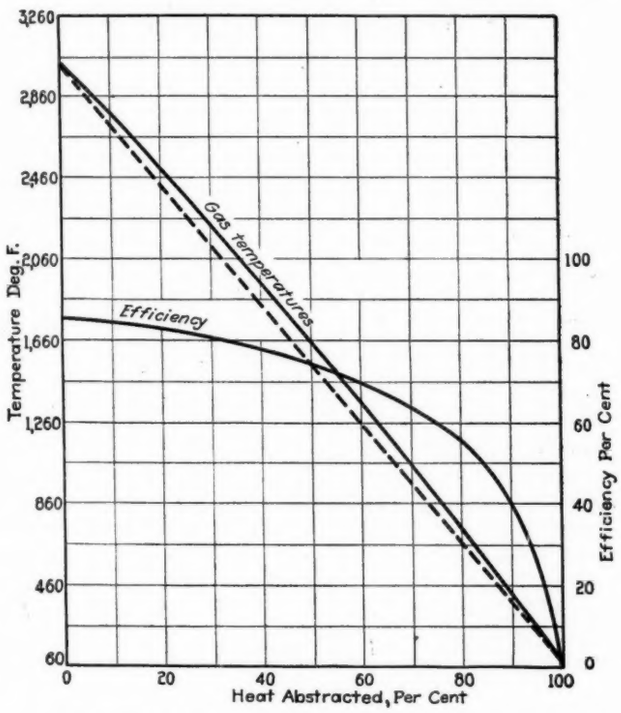


Fig. 1—Maximum possible efficiency of conversion of heat into work

$= 4.33$ B.t.u. The rise in temperature resulting from the combustion is equal to the heat of combustion divided by the heat required to raise the products of combustion one degree, or 14,500 divided by $4.33 =$ 3,350 deg. F. If the air enters at 60 deg. F., the furnace temperature will be $3,350 + 60 = 3,410$ deg. F., or $3,410 + 460 = 3,870$ deg. absolute. If the air supply is diminished, the combustion temperature will increase, and vice versa.

The problem of finding the maximum work that can be obtained from a pound of fuel burned in a furnace is the problem of finding the maximum work that can be obtained from a given weight of products of combustion at a given temperature; for the conditions stated, it is for 18.28 lb. of products at 3,410 deg. F. From the

Carnot principle this depends on the possible temperature of rejection of heat, which in this case is 60 deg. F. Imagine one B.t.u. taken from the gases. The maximum work obtainable from this heat is $\frac{t_1 - t_2}{T_1} \times 1 = \frac{3,410 - 60}{3,410 + 460} = 0.865$ B.t.u. = 673 ft.-lb.

But by taking away one B.t.u., the temperature of the gases has been decreased; as it required 4.33 B.t.u. to

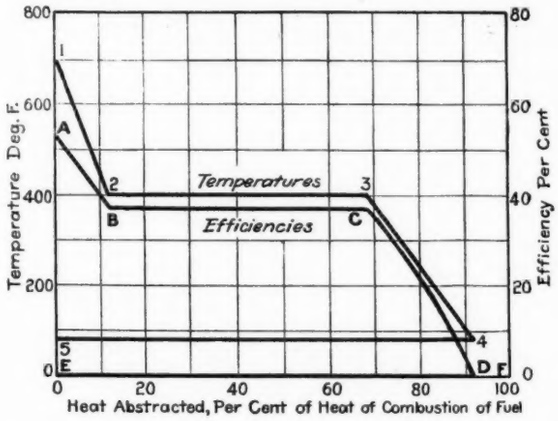


Fig. 2—Temperatures of heat intake and maximum possible efficiencies in steam cycle

raise the temperature of the gases one degree, the fall in temperature will be $1 \div 4.33 = 0.23$ deg. Consequently, the gases are now at a temperature of 3,410 — 0.23 = 3,409.77 deg. F. and the possible efficiency of conversion of the next B.t.u. into work is (3,409.77 — 60) ÷ (3,409.77 + 460), which is a little less than the value 0.865 just given. This diminution in the efficiency of conversion goes on continuously as the temperature of the gas falls till, when the gas is at 61 deg. F., the efficiency becomes

$$\frac{61 - 60}{61 + 460} = 0.0019.$$

To find the total work possible, recourse must be had to the calculus, which yields the equation,

$$\text{Efficiency} = 1 - \frac{T_2}{T_1 - T_2} \log_e \frac{T_1}{T_2}$$

The quantity $\log_e \frac{T_1}{T_2}$ is the hyperbolic logarithm of $\frac{T_1}{T_2}$, and its value can be obtained from tables such as those in Marks' "Handbook," page 58. For an air temperature of 60 deg. F., or $T_2 = 60 + 460 = 520$ deg. absolute, the possible efficiencies are as follows:

T_1, deg. abs	3,000	3,500	4,000	4,500
Efficiency	0.632	0.668	0.694	0.719

Maximum Efficiency for Any Cycle

The preceding theory and calculations are based on the assumption that the specific heat of the products of combustion is constant at all temperatures. This is not the case; it increases with increase of temperature. The specific heat of nitrogen varies from 0.247 at atmospheric temperature to 0.322 at 4,000 deg. F.; similarly, that of CO_2 varies from 0.210 at atmospheric temperature to 0.313 at 4,000 deg. F. In consequence of this variation the abstraction of a given amount of heat from the combustion gases will not produce so great a fall in temperature when the gases are hot as when they are colder. In Fig. 1 is plotted a curve showing how temperature falls with heat abstraction when T_1 and T_2 are equal to 3,520 and 520 deg. absolute, respectively. The broken straight line shows what would happen if the specific heats were constant.

At the initial temperature of 3,520 deg. absolute the possible efficiency is $\frac{3,520 - 520}{3,520} = 0.852$; after 10 per cent of the heat has been abstracted, the temperature is 3,250 deg. absolute, and the possible efficiency is $\frac{3,250 - 520}{3,250} = 0.84$. Calculating similarly for other temperatures and plotting the calculated efficiencies against the heat abstraction, the efficiency curve of Fig. 1 is obtained. Measuring the area under this curve and dividing by the horizontal length gives the mean height of the curve, which in this case is found to correspond to an efficiency of 0.692. That is, for a given furnace temperature of 3,060 deg. F., and with heat rejection at 60 deg. F., the maximum possible efficiency of conversion of heat into work is 69.2 per cent; the calculation with specific heat assumed constant gave about 66.8 per cent.

Maximum Efficiency for Assumed Cycle

A similar analysis of the heat reception in a proposed heat-engine cycle will permit the determination of the possible efficiency. Consider, for example, the case of a steam turbine taking in steam at 250 lb. absolute, superheated to 700 deg. F., and exhausting at 0.5 lb. abs., or 80 deg. F. The gases may be supposed to leave the economizer at 350 deg. F. and then to be discharged to the atmosphere. From Fig. 1, with a furnace temperature of 3,060 deg. F., the gases will reach a temperature of 350 deg. F. after 92 per cent of their heat has been abstracted, or the possible boiler efficiency in this case is 92 per cent.

For the condition stated, the highest temperature at

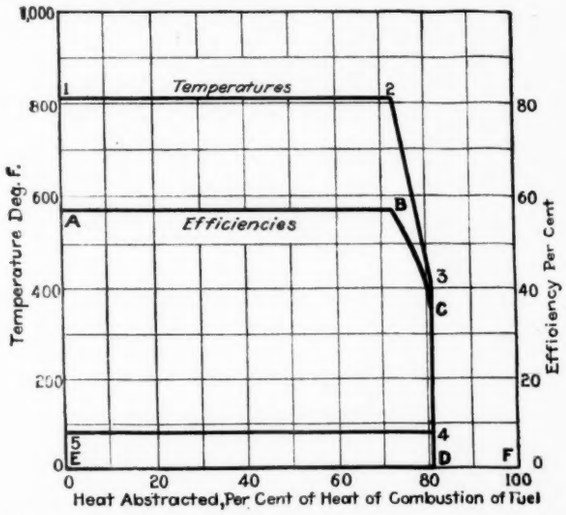


Fig. 3—Temperatures of heat intake and maximum possible efficiencies in mercury-vapor cycle No. 1

which heat is taken in is 700 deg. F. In Fig. 2 the line 12 shows the process in the superheater, the point 2 being at the saturation temperature of 401 deg. F. The line 23 shows the heat taken into the boiler at the constant temperature of 401 deg. F. The line 34 shows the heat going into the economizer and terminates at the point 4 at the condenser temperature of 80 deg. F.

From the steam tables the amounts of heat taken in —in these three places—are 161.5, 826.3 and 327.2

B.t.u., respectively, and as these total 92 per cent of the heat of combustion of the fuel, they are respectively 11.3, 57.8 and 22.9 of the heat of combustion of the fuel. The line 1234 represents the temperatures T_1, of reception of heat; the line 45 is the temperature T_2 of rejection of heat. Calculating the Carnot efficiencies and plotting, the line *ABCD* is obtained. The area *ABCDE* divided by the length *ED* gives the maximum possible cycle efficiency, which in this case is 35 per cent. The area *ABCDE* divided by the length *EF* gives the maximum possible plant efficiency, which must be $0.92 \times 35 = 32.3$ per cent.

The preceding method of analysis has no special advantage over direct calculation when applied to comparatively simple cycles such as that just considered. It is useful, however, in more complex cycles such as those used with the mercury-vapor process. In this case high cycle efficiencies can be shown, but they are likely to be associated with low boiler efficiencies and consequently with moderate plant efficiencies.

Method Applied to Steam-Mercury Cycle

Consider two cases: (1) With the combustion gases used only for heating and evaporating the liquid mercury; and (2) with the gases used also for superheating the steam and for heating feed water. With a furnace temperature of 3,000 deg. F., Emmet gives an exit gas temperature of 700 deg. F. for the first case and 425 deg. F. for the second. The possible boiler efficiencies from Fig. 1 are approximately 81 per cent and 90 per cent, respectively, for these two cases. With mercury vapor at 50 lb. abs. (812 deg. F.) and mercury condenser pressure at 0.5 lb. abs. (413 deg. F.), the heat supplied to the liquid is 13.4 B.t.u. per lb., and for

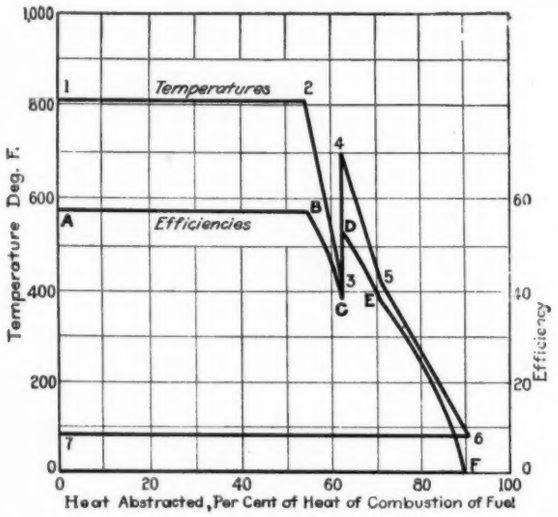


Fig. 4—Temperatures of heat intake and maximum possible efficiencies in mercury-vapor cycle, No. 2

evaporation 125.8 B.t.u. per lb. These correspond to the lines 32 and 21 in Fig. 3 and show a heat supply in evaporating the mercury of 73.2 per cent of the heat of combustion of the fuel, and in heating the liquid mercury, 7.8 per cent.

If the steam condenser pressure is 0.5 lb. abs. (80 deg. F.), the possible efficiency with this method of heat reception is obtainable from the efficiency line *ABC*. The area *ABCDE* divided by the length *ED* gives the possible cycle efficiency; divided by the length *EF*, it gives the possible plant efficiency. The actual values for this case are 58 and 47 per cent, respectively.

If the flue gases are used for superheating steam and for heating feed water, the percentages of the heat supplied at the various temperatures are obtained as follows: One pound of dry and saturated mercury vapor expanding from 50 lb. to 0.5 lb. abs. has a terminal quality equal to 0.78 and a latent heat of 0.78

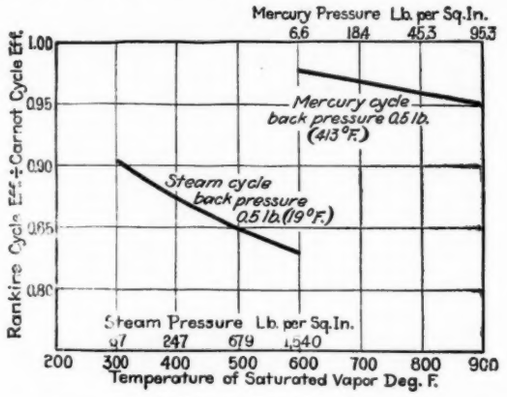


Fig. 5—Ratio of Rankine to Carnot cycle efficiencies for steam and for mercury

$\times 129.3 = 101$ B.t.u. At 413 deg. F. (285 lb. steam pressure) the latent heat of steam is 815.7 B.t.u. Consequently, the weight of mercury vapor used per pound of steam generated is $815.7 \div 101 = 8.08$ lb. With the steam superheated to 700 deg. F., the heat absorbed in the superheater per pound of mercury evaporated is $157.7 \div 8.08 = 19.5$ B.t.u. The heat supplied to the feed water per pound of mercury evaporated is $(387.7 - 48) \div 8.08 = 42$ B.t.u.

The heat absorptions are consequently in the following ratios: For evaporating mercury, 125.8; heating liquid mercury, 13.4; superheating steam, 19.5; heating feed water, 42. As the maximum possible boiler efficiency in this case has been seen to be 90 per cent, the heat absorptions expressed as a percentage of the heat of combustion of the fuel are 56.2, 6.1, 8.8 and 18.9 per cent, respectively. This heat supply is shown in Fig. 4 with the corresponding maximum possible efficiencies. The cycle efficiency is 50 per cent, the plant efficiency 45 per cent.

Heat of Liquid Less Important with Mercury

It should be observed that these maximum efficiencies are based on Carnot cycle operation. In steam engines and turbines the Rankine cycle is actually used. This cycle has an efficiency less than that of the Carnot cycle, but the difference between the two efficiencies depends on the physical properties of the working fluid. In the Carnot cycle all the heat is taken in at the highest temperature and is, therefore, used only for evaporating a fluid at a constant pressure when the working substance is a liquid or its vapor. In the Rankine cycle heat is also used for heating the liquid from the condenser to the boiler temperature. The Rankine-cycle efficiency will be closest to the Carnot-cycle efficiency when the heat of the liquid has the smallest ratio to the heat of evaporation, or when the ratio of specific heat of the liquid to latent heat is smallest.

The specific heat of liquid mercury is about $\frac{1}{30}$ that of water; the latent heat is about $\frac{1}{3}$. The ratio of specific heat to latent heat for mercury is consequently about $\frac{1}{30}$ that for steam; consequently, the Rankine-

cycle efficiency of a mercury engine is much closer to the Carnot-cycle efficiency than is the case with steam. In Fig. 5 is shown the ratio of Rankine-cycle efficiency to Carnot-cycle efficiency for these two fluids. It will be seen that whereas the Rankine-cycle efficiency with high-pressure steam is about 85 per cent of the corresponding Carnot-cycle efficiency, with mercury vapor it is over 95 per cent. This gives an initial advantage to mercury over steam of about 11 per cent.

Regenerative Cycle

The efficiency of the Rankine cycle can be raised to that of the Carnot cycle if the feed water is heated to the boiler temperature, not by heat from the outside but by the use of steam bled or abstracted from the engine or turbine during its expansion and used in a succession of feed-water heaters. This is called "regenerative" feed heating. With mercury vapor regenerative heating would not be of much value, because the Rankine-cycle efficiency is already nearly equal to the Carnot-cycle efficiency, but with steam the matter is important.

From the preceding considerations it may be seen that with mercury vapor used in a Rankine cycle, or with saturated steam used in a regenerative cycle, the cycle efficiencies can be made to approximate very closely to Carnot-cycle efficiencies, and therefore the use of Carnot efficiencies in Figs. 2, 3 and 4 is justified.

The standard practice in stating the performance of a heat engine is to give the "engine efficiency" or "efficiency ratio," which is the ratio between the heat input per unit of output for the ideal cycle and the heat input per unit of output for the actual engine. This is the same as the ratio of the thermal efficiency of the actual engine to the thermal efficiency of the ideal engine. The Rankine cycle is no longer suitable as an ideal for steam plants because of the use of regenerative heating. The only suitable ideal is one in which the actual temperatures of heat intake into the cycle are taken into account, for upon those temperatures depends the possible work output.

Application to Internal-Combustion Engines

Similar methods can be applied to internal-combustion engines. In these heat is supplied at all temperatures from that at the beginning of combustion (or explosion) to that at the end of combustion; the possible work output depends on the amount of heat taken in at each temperature. There is need here of defining the temperature of rejection of heat. If it is taken as the temperature of the atmosphere, it would require expansion to pressures well below atmospheric pressure, which would demand the use of a compressor to discharge the products of combustion to the atmosphere and would as a consequence result in a net loss of work.

The natural standard condition with gases is exhaust at atmospheric pressure and this gives a different minimum temperature for the Carnot cycle corresponding to each infinitesimal heat addition. The methods given earlier in this article can be used and will give the maximum possible cycle efficiency. In internal-combustion engines cycle efficiency and plant efficiency are identical except in the case where the fuel undergoes preliminary modification (as in a gas producer) before being taken into the engine.

Inspection of Electric Elevators

By WARREN HILLEARY
Superintendent, The Royal Indemnity Company, New York City

WHEN the car is at the bottom landing, there should be at least one and one-half and preferably two, full turns of hoisting cable on the drum, and when at the top landing there should be a like number of turns of drum counterweight cable on the drum. The car-counterweight cables generally can be examined from the inspector's position on the floor at the overhead works. They may be continued in service even when in worse condition than either the hoisting or the drum counterweight cables, for as a rule the car counterweights are light as compared with the drum counterweights, although the cables are customarily of the same size. In every instance most minute examination of the hoisting cables should be made at the point where they enter the bell socket or other fastening device on the car or counterweights. With an overhead type of machine there is a considerable twisting of the cables at the point where they connect to the car, and the tendency is for the cables to twist off or crystallize. Crystallization, of course, cannot be detected, but there are instances where the inspector found broken wires at the point where they entered the bell clamp. If even one wire in a cable is found broken at the fastening, and if the cables have been on for more than three or four months, it is advisable to have them cut off and rebabbitted. The danger may be greater than is evident.

The governor should be set and released four or five times. The inspector should see that the potential or main-line switch is open, so that the car cannot be started while the governor is set, and especial care should be exercised to make certain that the governor is released before the car is started downward. When the eccentric jaws are brought together by the action of lifting the governor weights, care should be taken to see that they are gripping the safety cable and that there is at least $\frac{1}{8}$ in. clearance between the flanges of these eccentrics. There are numerous instances where the flanges came in contact with each other before the cable was gripped in the V-grooves.

All bolts and nuts on the machine should be examined in an effort to learn whether they are loose and if any cotter pins are missing. The switchboard should be examined for burned contacts, burned places on the slate or marble, between parts carrying differences of potential. Testing the slack-cable device on such machines as have this device and also a potential switch, will determine the adequacy of the slack-cable device and the potential switch at the same time.

On a polyphase alternating-current machine there should be a reverse-phase relay. An alternating-current machine without a reverse-phase relay is unsafe, since a transposing of the lines in the street or elsewhere may have the effect of reversing the motor, which would constitute a dangerous condition. Direct-current ma-

chines need no reverse-current relay, since the transposing of the leads going to the motor will not reverse the direction of rotation of the armature.

When the worm-gear case is open and the gear in motion, it should be observed whether the gear is well submerged in oil and the surfaces of the gear teeth in good condition. If the appearance of the oil indicates that it is not in good condition, the worm-gear case should be thoroughly washed out with kerosene, using at least five gallons for the purpose, after which fresh gear oil is put in the case.

Where considerable cutting of the gear is evident, through fine particles of bronze in the oil, about two pounds of sulphur should be put into the gear oil. This will frequently reduce cutting and sometimes stop it entirely. The bronze gear teeth will seldom be found worn to a dangerous extent, and even though they may be worn to almost a sharp edge, it does not follow that the condition is dangerous, but a new gear and new worm should be installed. A new worm may not always be necessary when a new gear is installed, but generally it is advisable to renew both in order that there may be a proper fit between the two.

Some types of machines have ball thrust bearings, others the roller type and others the disk type. These bearings are not always adjustable, and not infrequently they will be worn or contain a broken ball or roller. The evidence of worn or disarranged parts is a peculiar grinding noise, which may be evident when the car is going upward and not when it is going downward, or vice versa. It is difficult to determine from the noise just what may be wrong with the thrust bearings. The exact nature of the repair cannot be determined without opening the bearing.

Packing Gland on Wormshaft

See that the packing gland on the wormshaft is filled with packing of a kind and in a manner that will prevent the oil from leaking out of the gear case and yet will not create excessive friction on the shaft, since excessive friction will cause heating and grooving of the shaft. Where the packing has not been changed for a long time, as is frequently the case, it becomes non-resilient and vibration backs the nuts off the gland studs. Flax or hempen packing is more suitable than any combination of rubber and other materials, because oil will not readily affect them. A properly packed gland does not require the use of a wrench to tighten the packing and should allow only a small quantity of oil to leak out in a day's running.

Unless the inspector understands the method of setting and releasing the slack or broken-cable type of safety, operating the machine in the downward direction to get slack in the cables after the car has been blocked, and the placing of the tension on the cables after the test, he should not attempt to make this kind of test, because he may either seriously injure himself or get the cables off the winding drum and be unable to get them back in position. The slack- or broken-cable safety device can be in perfect condition and should the cables break on the hoisting drum of the machine, the car can go down to the bottom of the elevator shaft without the safeties gripping the guide rails. This is due to the friction of the cable on the overhead sheaves, and the weight of the cable on the drum side being sufficient to hold the spring, so the jaws of the safeties will not touch the guides.

As a general rule, on drum machines other than overhead the drum-counterweight cables will be found to be worn and broken most on the side that does not touch the drum. This is because the idler has a greater wearing and breaking effect on the cables than the drums, owing to the smaller arc of contact.

The dynamic brake on an electric elevator cannot be tested while the car is standing still. This test is accomplished while the car is being brought to the landings in the usual manner, so there is no reason to test the dynamic brake by any means other than the normal starting and stopping. The mechanical brake is tested in the same manner, since, obviously, if the car is brought to rest at the floor without undue sliding and if the slide at each floor is approximately the same with the same load and speed, then no other test of the mechanical brake need be made. However, the inspector should see that the rivets that hold the brake linings to the castings are not in contact with the pulley, and that all cotter pins, nuts, locknuts, springs, etc., are in proper position.

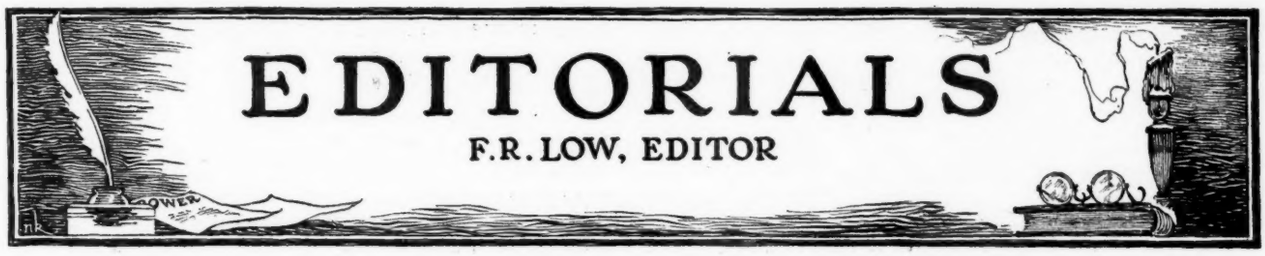
Making Brake Adjustments

To operate properly on any type of electric elevator, the friction brake, whether it be electrical or mechanical, should be so adjusted that it will release a fraction of a second after the starting switch closes. If the brake releases before the starting switch closes, an under-counterweighted car and load may start downward even though the lever be on the "up" position. If the car be over-counterweighted, it may start upward even though the lever is on the down position. Theoretically, the car will not start on a worm-gear type of machine until the motor starts, but in actual practice the reverse is not infrequently found to be the case. If the brake opens too long after the closing of the starting switch, the car will start too abruptly. Either condition is dangerous to the cables.

Inspection of Controller

Where the shunt field of a direct-current motor is open when the starting switch is open, inspectors must be certain that the contactors for the shunt-field connection are in good condition, for it is dangerous to have the machine operate without the shunt-field circuit closed. Where the shunt field remains closed when the machine is stopped in some cases, an external resistance is introduced in this circuit regulated by either a mechanical or magnet switch. These switches must be in good order and properly timed, so that the introduction or elimination of the external resistance will occur at the right instant. Otherwise, improper starts and stops of the car must result. Where acceleration switches are used, inspectors must be certain that they close in proper relation to each other, since if the high-speed switch closes first, it will cause a dangerously quick start. Practically all electrical difficulties in connection with friction brakes and external resistances in either the armature or shunt-field connections are communicated to the hoisting cables, in the form of a sudden strain, therefore the electrical equipment must be in good condition and properly timed.

The hatchway limit stops, if any, should be examined and tested while the car is standing. The test on most types will consist of opening the switch by hand, and this, in turn, should cause the potential switch on the control board to open. Each should be tested in turn and each should have proper effect on the potential switch.



The "Fifth Estate"

IN OLD England there were three "estates" or classes —the lords spiritual, the lords temporal and the commoners—until Edmund Burke pointed out a "fourth estate"—the press—whose members had a certain power belonging to none of the other three. Now a "fifth estate" has been discovered by Arthur D. Little, of Cambridge, Mass. According to Doctor Little, who recently voiced his sentiments at the Franklin Institute centenary in Philadelphia, the "fifth estate" is "that small company upon whose creative effort the world depends for the advancement of science." He urged that the knowledge, vision and open mind possessed by the members of the fifth estate fitted them above all other men to take a leading part in solving the problems of government and formulating national policies. "Could the springs of human conduct and the affairs of peoples now be regulated only as wisely as we know how, there would be work and leisure and decent living for all."

Here is much food for thought. We all know that bungling is still the principal art practiced by mankind. In every line of work there are certain established truths that are well known to a few, but have never percolated to the mass of men. Underlying all business and trade are the principles of economics, yet the average small business man and tradesman knows practically nothing of the great tides that are shaping his own little whirlpool of activity. Government policies on such matters as public health, employment, water-power development, the conservation of resources, development of harbors and rivers, etc., are largely shaped by politicians who know little and care less about the fundamental principles involved and the effect of their actions upon the present and future welfare of the nation. Expert disinterested knowledge does seep here and there, but too often the scientist and engineer play a secondary rôle in working out the details after the main pattern has been determined by the less competent but more vociferous brothers of the "second estate"—"the lords temporal." Talk still has the edge on deeds.

The bright side of the picture is that the fifth estate is growing in power little by little. Now and then the brilliant scientist is pulled out of his cubbyhole to pronounce an opinion listened to with some respect. Here and there a real engineer is permitted to offer a few facts for the consumption of a people fed up with guesses and dreams. Slowly but surely the public is beginning to recognize the hand that feeds it.

Unskilled in and temperamentally unfitted for the arts of the demagogue or the wiles of the political jobber, the representatives of pure and applied science are beginning to be heard. Their works speak for them—great buildings, power plants, intricate machinery, marvelous transportation systems, ships, telephones, radio, modern surgery, airplanes, wonderfully correlated manufacturing processes.

As these works are multiplied and perfected, the public will more and more look to the "fifth estate" to build them enduring foundations in the political swamp.

A Substitute for Rankine Efficiency

THERE are many efficiencies. Probably the most important is the "thermal efficiency," the ratio of useful energy output to energy input. This efficiency is usually rather low in steam plants—only the exceptional plant reaches twenty per cent—so engineers need some other term for efficiency which will express the performance of the plant as a fraction of what is ideally possible. Up to recent times the generally recognized standard for this purpose has been the Rankine cycle with complete expansion. The efficiency of this Rankine cycle is the same as the thermal efficiency of a plant operating with the same initial steam pressure and quality and the same back pressure, and consisting of an engine or turbine expanding the steam adiabatically to the back pressure, a heater in which the feed water is brought to the exhaust temperature, and a boiler of one hundred per cent efficiency. The ratio of the actual thermal efficiency of a plant or engine to that so computed is termed the "Rankine efficiency," or "efficiency ratio."

This was not an unreasonable standard in the early days of steam engineering, but recent progress has reduced its significance. Bleeder heating, in particular, has given a blow to this standard, since with the regenerative cycle the limiting efficiency of a steam plant is that of the Carnot cycle.

In an article in this issue Lionel Marks, professor of mechanical engineering at Harvard and editor of Marks' "Handbook," proposes an entirely different standard. He goes back to the boiler furnace and assumes a fuel of definite composition to be perfectly burned with a definite percentage of excess air, which in turn produces a definite weight of gas of definite composition and at a definite temperature, assuming no heat losses in its production. Professor Marks then proceeds to apply the Carnot cycle step by step as the gases are cooled down to room temperature, which gives the maximum thermal efficiency obtainable with any kind of power-producing apparatus furnished with the given heated gas. This ultimate standard is very high, averaging around seventy per cent.

The next step is to compute, by a similar step-by-step method, the maximum efficiency theoretically obtainable with any given plant cycle assuming a definite steam pressure and temperature, exhaust pressure, method of heating the feed water, etc. The method is applicable to mercury and binary-vapor systems as well as to steam plants.

In the case of steam plants the method leads to two

theoretical efficiencies—the maximum possible efficiency of any plant with the given fuel, excess air and heat rejection temperature, and the maximum possible efficiency of any engine with the given conditions of taking in and rejecting heat. From these are immediately deducible the actual efficiency ratios of the engine and plant compared with the maximum theoretical engine and plant efficiencies respectively.

Professor Marks' method offers a definite and standard means of comparing various cycles. Its use should be particularly advantageous in estimating the gains from bleeder heating, air preheating, various mercury-steam cycles, etc.

Design of Safety Device Depends on Operator and Inspector

A MECHANISM designed to operate when a machine element or human being fails to function in the prescribed manner, may pay for itself many times over in a single day. Safety devices designed in the drafting room from purely engineering viewpoints have in some instances proved markedly inferior in practice to those made up on the job. It appears that the intelligence and attitude of both operator and inspector are often factors more vital to satisfactory performance than purely engineering considerations alone.

Switches for preventing overtravel of crane hoists, which were subject to discussion at the recent convention of the Association of Iron and Steel Electrical Engineers, as described in the September 23 issue, illustrate well these principles. Apparently, in some instances less damage and better satisfaction resulted by removing the hoist-limit switches altogether and placing the entire responsibility on the operators. It is evident that a sufficiently high grade of operator must have been available so that this plan could succeed, and it is not to be expected that such a favorable condition would hold generally.

A limit switch that functions to stop the hoist close to the highest allowable position, but yet permits it to be lowered readily, has sometimes been found unsatisfactory, owing to the craneman often depending on it to stop the hoist rather than shutting off the hand controller in time. The limit switch, like other control elements, sometimes may itself fail. Not only is the likelihood of damage increased when the device is regularly utilized, but also maintenance and repairs become more frequent and costly.

One means of combating this condition is to design the switches as "non-resetting," or in other words making it necessary for the craneman himself to adjust the limiting device after it functions, and go to an appreciable inconvenience before the hoist can again be put in service. It will then be more convenient to utilize the controller instead of depending on the switch. An objection to this plan is the loss of time in resetting, perhaps in important work like pouring a ladle of molten metal. Another is that it provides an incentive for the craneman to render this limit inoperative, where the condition then reverts to that previously mentioned, except that it is produced surreptitiously.

Another way to safeguard the switch performance is to provide a telltale arrangement whereby a record is made of when and by whom the switch is tripped so that a suitable penalty can be imposed. The success of this plan, however, is dependent on inspection and other circumstances. Another instance may be cited with profit, wherein some limit switches made of ordinary lumber and scrap copper, costing an insignificant amount, completely solved the problem where unreliable labor was used. These were in plain view, and if deliberately put out of commission, the condition could be detected ordinarily by observation. Inconvenience in resetting prevented their being utilized in ordinary operation.

It is evident that with design, inspection and maintenance of a sufficiently high order, there would be no more probability of accidents where the operator continually utilized a well-designed limit switch instead of the hand controller, than that of any other element of automatic electric control which is in constant use. In other words, sufficiently expensive design and maintenance could compensate for improper use by the operator. Commercial costs, however, and practical possibilities of trouble, would hardly recommend such a condition.

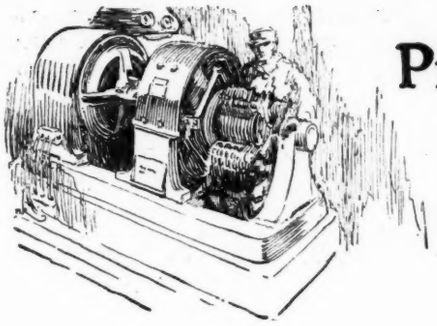
Both the personal element and the engineering features are to some extent under the control of the executive directly responsible for the department where a safety device is in use. While in standardizing safety devices of a given type, it is necessary to consider a diversity of circumstances, there remains, however, the possibility of altering conditions of the engineering or human element somewhat in conforming to desired standards.

Improvements in Hydro-Electric Machinery

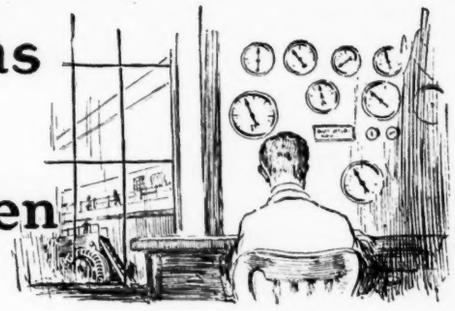
THE leading article in this issue describes the Holtwood Plant of the Pennsylvania Water & Power Company. There are many interesting features in this plant, but those that indicate the improvement in design of hydro-electric equipment in the last twenty years are of equal importance to any others. The first seven units installed were of the double-runner vertical-shaft type discharging into bent draft tubes. This design, although a forerunner of the single-runner vertical-shaft efficient unit of today, was of comparatively low efficiency due to the draft-tube arrangement and was difficult to maintain because all operating parts were submerged.

In the new units with single runners the horsepower has been increased about twenty-five per cent and the efficiency nine per cent over that of the first units installed. On account of the increase in specific speed of the units the energy to be regained in the draft tube has increased from about seventeen per cent of the total head on the turbines for the older machines to nearly forty-five per cent for the latest units, which shows the importance of draft-tube design in the modern high-speed turbines used in low- and medium-head installations.

The improvement in design is not all confined to the hydraulic units, as in this plant it is also reflected in the redesign of some of the older generators. This is well illustrated in the rewinding of No. 3 generator for a sixty-five per cent increase in capacity, using the same stator frame and core and changing the winding only. In this case the turbine and shaft had sufficient capacity to take care of the additional load. Such improvements in design as these make reduced rates possible notwithstanding the increased cost of labor and materials that go into the construction and operation of power stations.



Practical Ideas *from* Practical Men



Furnace Linings and Arches

In burning 1 lb. of pure carbon to CO_2 using 17.8 lb. of air for combustion, you have about 13 per cent CO_2 in the gases, 18.8 lb. of gas per pound of carbon consumed, and an average temperature (neglecting losses by convection in the furnace walls) of 3,000 deg. F. above the temperature of the air entering the furnace, at the point where combustion is completed and gases thoroughly mixed.

There are plenty of good firebrick that will resist this temperature indefinitely without fusing; but under actual furnace conditions combustion is in progress at this point, and as the gases are not completely mixed, it follows that strata of gases in the furnace may vary in temperature from a low point to 3,750 deg. F., resulting in an average temperature of 3,000 deg. F. above atmosphere when completely mixed.

In other words, only 11.6 lb. of air is used per pound of carbon when burned, regardless of the percentage of excess air present. Therefore firebrick arches and side walls burn away when these flames come in contact with them.

It has been customary for many years, for basic reasons, to build side walls with a removable firebrick lining, making it unnecessary to renew the entire wall when repairing. Refinements in practice in late years have provided for side-wall linings that are easily replaced in sections of about six courses in height and are supported by tile extending into the wall.

This construction accomplishes three objects: Repairs are easily and cheaply made; the pressure on the lower courses due to the weight of the brick above is greatly reduced, thereby insuring longer life to the brickwork; this lining can be air-cooled to some extent, as the air is at all times under control of the operator, without detriment to furnace efficiency.

Side walls built in this manner need no cement-gunning except in emergency cases, as their endurance makes them cheaper in the long run than the materials and labor required for building them up with ganister, etc., in the majority of cases.

With regard to arches, the latest practice appears to be the double suspension, with a permanent upper arch, to which tile is suspended to provide an easily renewable wearing surface. The lower tile fit closely and no fireclay is used. A space between upper and lower tile allows a small volume of air to be introduced, thereby not only slightly lowering the temperature of this tile, but also preventing actual contact of the destructive gases to some extent, and as only a small amount of air is required for this purpose, it is really a benefit to a heavily fired furnace.

I read the article by G. B. Randall in the July 22 issue with much interest. He seems to have given considerable thought to this subject. But in my opinion the article shows the general tendency to oppose radical improvements by inventing means to get results with inferior construction.

J. W. FOWLER.

Cleveland, Ohio.

Sampling Boiler Water from the Water Column

Usually, in sampling water from boilers for testing, the water samples are taken from the blow-down lines close to the mud drum. These samples will give the maximum tests for alkalinity, solids, etc. But it often happens that no provisions have been made for sampling water at this point, and the most convenient place for sampling is the water column.

This situation arose in one of our boiler houses, and samples were taken from the water column without blowing it down first, and for a long time we were at a loss to understand why this boiler house tested so low. Using the alkalinity test alone, the results were as follows:

	Parts per Million	
No. 1 boiler house (one boiler)	80	Water column
No. 2 boiler house (one boiler)	250	Mud drum blow-down
No. 3 boiler house (one boiler)	300	Mud drum blow-down
Feed-water heater water.....	50	

We increased the blow-downs on the No. 2 and No. 3 boiler houses and decreased that of No. 1, without much change, until it dawned upon us that we were sampling distilled water in the water columns of the No. 1. The water column had been acting as a condenser of steam, and there was not enough circulation to keep the water stirred, so that the true water would drain back to the boilers and the water in the column would become purer all the time. The other boiler houses had provisions for sampling from the blowdown. At times the samples from the water column at No. 1 boiler house would show lower alkalinity than the feed-water heater.

We then proceeded to blow the water column down well before sampling, and the following analyses are given to show the difference between the two methods. The boilers were blown down more than a gage, and the water column was blown down thoroughly, before the second samples were taken.

	Before Blowing Down, Parts per Million	After Blowing Down, Parts per Million
Water-column samples, No. 3 boiler house	192	484
Blow-down line from mud drum.....	548	540
Feed-water-heater water	65	65

There was practically no change in the blow-down water before and after blowing down.

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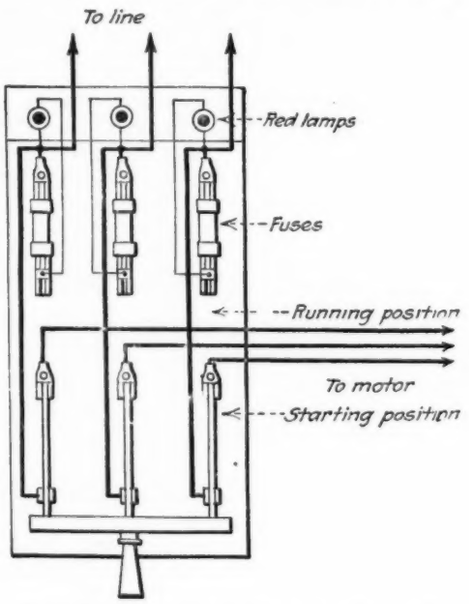
Signal for Indicating When a Fuse Blows on Induction Motors

A polyphase motor will not start on single phase, but will continue to run and carry its load if brought up to speed on a polyphase circuit. For example, if, after starting a three-phase motor and bringing it up to speed on a three-phase circuit, one of the fuses was removed, the motor would be operating on single-phase current.



Signal lamps' connections to show when fuses are blown

Under such condition the active phase would take an increased current to carry the load. If the motor was operating at over 50 per cent rated load, the current taken by the single phase would be sufficient to cause serious heating, and the motor might be burned out before the trouble was discovered.

Where the fuses on a polyphase alternating-current motor are cut out of circuit during the starting period and switched back in again after the motor has come up to speed, there is always danger that the machine may be operated single-phase due to blowing of one of the fuses. The arrangement in the figure shows a usual one for small-sized motors. When the switch is closed to the starting position, the motor is connected directly to the line and will be supplied with power on all phases so that it will come up to speed. When the switch is thrown to the running position, if one of the fuses were blown, the motor would continue to operate single-phase as explained in the foregoing.

To give the attendant an indication of the condition of the fuses, I connected three red lamps to the fuses as shown. As long as the fuses are not blown, the lamps will be out when the motor is in service. However, should one of the fuses blow, the lamp connected to this fuse will be in series with the motor and will light, showing which fuse has failed. The lamps, being red, are more liable to call the attendant's attention to the condition.

This plan was first applied to a safety type of starting switch, which, when thrown to the starting position, closed two switches. One of these connected the motor directly to the line, and the other short-circuited the fuses. In the running position the switch that short-circuited the fuses, while starting, opens and puts the fuses in circuit. Trouble had been experienced from one of the fuses opening and operating the motor single-phase, thus endangering the windings. Since this signal system was installed, the defective fuses are detected immediately by the workmen, who stop the motor and have the fuse replaced before any damage is done to the motor.

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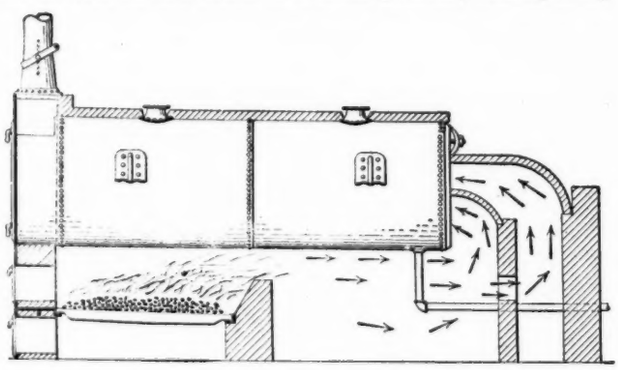
An Experiment in Boiler Settings

Some time ago I noticed in one of the engineering journals a peculiar boiler setting, which recalled an experiment with different types of setting I had seen about twenty-five years ago.

The engineer who planned the setting, in which a baffle was placed in the combustion chamber, as shown by the illustration, gave the following reason: That a vessel containing hot water had a higher temperature near the surface than at the bottom. The purpose of his setting was to direct the hottest gases through the lower tubes and the remainder through the upper rows of tubes, thereby directing the most intense heat where he considered the temperature of the water was the lowest. I was quite young at that time and never heard whether the experiment was of any advantage, and I am presenting it here for the discussion of *Power* readers.

The boiling point of water at atmospheric pressure is approximately 212 deg. F.; at 28 in. vacuum, approximately 100 deg. F.; under 100 lb. pressure, about 340 deg. F.; consequently the boiling point varies with the pressure in the boiler. When the water boils, small bubbles form, which rise to the surface and enter the steam space.

Owing to the weight of the water the pressure at the bottom of the boiler is a little greater, consequently



Baffle in combustion chamber to direct hottest gases through the lower tubes

the boiling point would be slightly higher at the bottom of the boiler than at the surface. The feed water entering the boiler at a lower temperature will also seek a low level in the drum.

In conclusion I wish to ask the question, Has an actual test ever been made to determine if there is any difference in the temperature of the water in a boiler at the surface of the water and at the bottom? If so, what did it show?

A. A. FETTE.
Cincinnati, Ohio.

Comments from Readers

Why Should the Smaller Pumps Give Better Results?

With reference to the inquiry of F. L. Pakes in the Aug. 26 issue, entitled "Why Should the Smaller Pumps Give Better Results," it is obvious that some error exists in the data submitted.

A 1¼-in. stream with a nozzle pressure of 75 lb. will deliver about 395 gal. per min. (pressure being measured by a pitot-tube gage), and the friction loss through the 2,000 ft. of 4-in. pipe would be nearly 100 lb. It is not likely that a centrifugal pump rated at 125 gal. per min. would deliver this amount of water even at the reduced pressure of 50 lb. unless supplied under a considerable pressure head.

If two streams, 1⅛ and 1¼ in., were maintained by the fire engine, as stated, at 75 lb. nozzle pressure the discharge would be approximately 320 + 395 = 715 gal. per min., obviously more than the centrifugal could deliver, and for this discharge the friction loss in the 2,000 ft. of 4-in. pipe would run up close to 300 lb., an impossible condition in the present stated case.

A duplex pump rated at 900 gal. per min. may or may not deliver that amount, depending on its operating speed. This speed is not given. Evidently, the pump did not deliver that quantity of water as the suction pressure at the fire engine fell to a vacuum of 7 inches.

Furthermore, the 4-in. pipe has a summit 500 ft. from the fire engine. This summit forms an air pocket where air will accumulate, and this air is not readily carried along by the water. Its displacement even in part will vary with the velocity of flow in the pipe. If this arrangement must be used, better results might be obtained by connecting an air vent at the high point, and bleed off the air, through a check, otherwise the accumulated air will act as an obstruction in the pipe, cut down its carrying capacity and build up pressure at the pumping-station end. If this summit is allowed to remain unvented, erratic performance of the pumps may be expected and poor service result.

In any event the size of the pipe and its length is against very good service from the combination.

Newtonville, Mass. EZRA E. CLARK.

Flow Meters for Refrigerating Plants

I was interested to note the editorial in the Sept. 2 issue of *Power* entitled, "Flow Meter Needed for Refrigerating Plant." This question has been a matter of consideration and discussion for several years by refrigerating engineers. It is possible that some form of flow meter could be designed similar to a flow meter for other vapors, but securing extreme accuracy with such a meter is next to impossible on account of the conditions of operation together with the properties of ammonia vapor. For example, the quality of the vapor drawn into the cylinders of the compressor is likely to change frequently, and it is difficult to determine how much liquid ammonia is mixed with the gas. The gas leaving the compressor is superheated, but the degree of superheat will change immediately on a change in the quality of the suction gas. Furthermore, it is difficult to determine accurately the mean temperature of the discharge gas, as it is frequently 200 deg. above the room temperature, and the largest part of this temperature increase is superheat.

By glancing at the ammonia tables, it will be observed that the specific volume of the vapor changes rapidly with slight changes in pressure, so that a slight error in the observation of suction pressure would cause a considerable error in calculating the flow if a meter was used on that side. For instance, at 0 lb. gage the volume of the vapor in cubic feet per pound is 18, and at 1 lb. gage it is 16.9, so that only one pound increase in pressure results in 6 per cent decrease in volume at that point. Suction pressures around 0 lb. gage are quite common in low-temperature work.

In metering the liquid from the liquid receivers, the errors due to operating conditions and the properties of the gas disappear. The temperature and pressure of the liquid may be accurately determined. A difference of five degrees in temperature of the liquid affects its density less than one per cent. It is easily possible to determine the temperature of the liquid within one degree and its temperature is not subject to a rapid change at any time.

The editorial speaks of the objection to the measurement of the liquid ammonia leaving the condenser on account of the tendency of the liquid to come down in slugs. As is intimated, this condition should not and does not exist in a properly designed plant; however, the meter should never be placed between the condenser and the liquid receiver. If the meter is placed in the line leading from the liquid receiver and the outlet of the liquid receiver is sealed, as it must be in any refrigerating system, slugs of liquid from the condenser will have no effect whatever on the operation of the liquid meter.

The editorial states that venturi meters have been pronounced inaccurate when the flow is variable. They are no more or no less inaccurate than such meters used for any other fluid with a variable flow, and it is well known that venturi meters and other types of fluid meters are regarded as accurate when the flow is variable.

It is true that a manometer or any other form of differential pressure gage only indicates the rate of flow at the time of observation, and for a record of the quantity of liquid handled over any period the most convenient form is a recording meter. Such meters for ammonia are precisely as accurate and satisfactory as any other recognized recording meters, and they are no longer in the experimental stage.

It is hardly necessary to point out that the principle of the venturi meter depends on certain well-known laws and that its accuracy has been firmly established. Aside from viscosity the nature of the liquid in no way changes the pressure-velocity relation. The viscosity of liquid ammonia is too low to intro-

duce any error in the formulas as derived from the flow of water. The only requirement is that the liquid shall not give up any vapor at the throat or point of lowest pressure. In the case of liquid ammonia this is readily accomplished by subcooling the liquid from the receiver below the saturation temperature existing in the throat of the meter or at the nozzle. It is quite evident from some experiments that have been made and published on venturi meters for ammonia that this requirement was not completely carried out. There are several venturi meters that have been in constant use for a number of years in which no difficulty is experienced whatever.

In the last paragraph of the editorial it is stated that "in the absence of instruments capable of measuring the work done in a plant, it is surprising how near to the maximum possible results many plants go in every-day operation." I do not see how it is possible to ascertain what the results of any plant are without a suitable means of determining the output of the plant, and it has been my experience that in most plants the performance is very much poorer than it is generally assumed to be in the absence of measuring devices.

On the whole I am firmly convinced that a liquid meter and not a gas-flow meter is the correct device for measurement of the output of the refrigerating plant.

New York City. GEORGE A. HORNE,
Merchant Refrigerating Co.

Single vs. Two-Pass Condensers

Mr. Bancel's article in the Sept. 2 issue, page 371, advocating single-pass condensers, is interesting as an elementary study in design, but, because the analysis is made to fit the author's required answer, it should be extended.

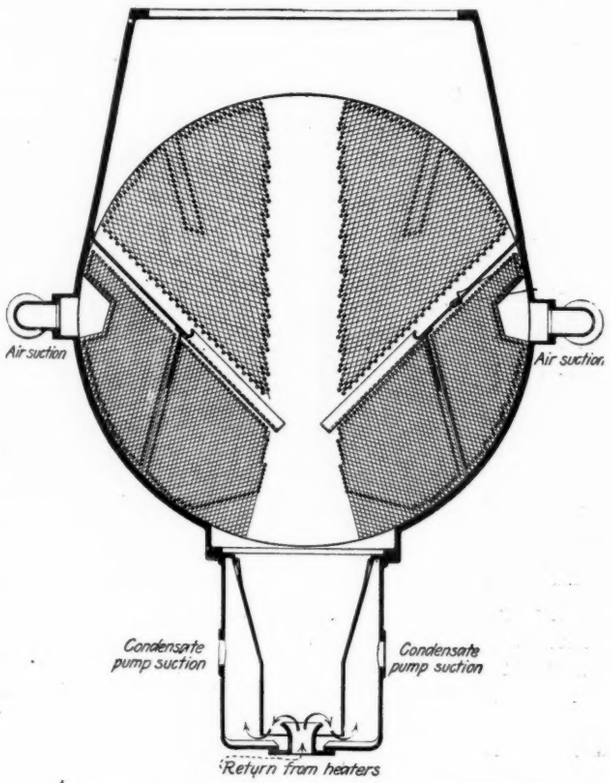
The illustration used, the condenser on the U. S. battleship "Maryland," is of a design that would not be given consideration for a modern power station. Furthermore, the article quoted from the A.S.N.E. Journal of November, 1922, was prepared to give a comparison between a good and a poor design and the poorest of nine tests on the two distinct designs has been taken.

TABLE I—COMPARATIVE DATA ON ¾- AND 1-IN. TUBES

Diam. O.D. In.	Cross-Section, Sq.In.	Inside Surface, Sq.In. per Inch Length	Inside Surface per Sq.In. of Cross-Section
¾	0.3339	0.1706	0.511
1	0.6410	0.2362	0.369

The condensing effect of tubes through which cold water is being circulated, is a direct function of the heat transfer in B.t.u. per square foot per degree mean temperature difference between the steam and the circulating water. The heat transfer, in turn, is a function of the tube cleanliness, freedom of non-conductive air envelopes on the steam side, the temperature of the water, and most important, the velocity of the water through the tubes. Cleanliness is usually disregarded in design as the tubes are assumed to be maintained commercially clean. Air removal is likewise disregarded, but should not be, as the tube layout and design have an important bearing on this factor.

Considering the effect of velocity and assuming the same amount of circulating water, there would be no difference between a single-pass condenser with 20-ft tubes and a two-pass condenser with 10-ft. tubes, either in velocity, power consumption or heat transfer, except to the disadvantage of the single-pass condenser, because the condensate would be several degrees colder. As Mr. Bancel has shown, it would be necessary to circulate twice as much water in a single-pass as in a two-pass condenser having the same tube size and length, to obtain the same velocity and heat transfer, and with increased pumping costs.



Two-pass condenser of 27,000-sq.ft. cooling surface

A corresponding velocity may be obtained with a single-pass condenser and equivalent water quantity by reducing the tube size, but this again increases the pumping costs owing to the higher friction loss in the smaller tubes. For example, with a velocity of 6 ft. per second, the friction loss per foot through a ¾-in. tube is 0.308 ft. and through a 1-in. tube, 0.230 ft. The smaller tube is not cleaned so easily and becomes fouled and stopped up by foreign matter more readily. Table I, applying to No. 18 B.w.g. tubes, gives the relative proportions of these two sizes of tubes.

Such comparisons of multi-pass condensers as are

TABLE II—RESULTS OBTAINED IN RECENT TEST OF TWO-PASS CONDENSER

Steam per Hour, Lb.	Temperatures, deg. F.					Pressures, in. Mercury				Water, G.P.M.	Ratio Water to Steam	Heat Transfer B.t.u. per Sq.Ft. per Deg. per Hr.	Per Cent Load
	Av. Steam Temp. T_s	Circulating Water In-T_1	Circulating Water Out-T_2	Terminal Diff. (T_s-T_2)	Condensate in Hotwell	Barometer	Mercury Column	Absolute Pressure	Vacuum Drop Through Condenser				
176,626	91.00	68.0	84.0	7.00	91.0	29.01	27.52	1.49	0.035	20,900	59.4	464	100
160,863	89.50	69.0	83.5	6.00	89.5	29.03	27.65	1.38	0.037	21,100	68.5	498	...
159,799	89.25	69.0	83.1	6.15	89.2	29.03	27.66	1.37	0.037	21,540	67.4	480	...
146,110	88.00	69.0	81.9	6.10	88.0	29.04	27.70	1.34	0.037	21,550	73.8	465	90
145,562	87.50	69.0	82.0	5.50	87.5	29.05	27.70	1.35	0.037	21,300	73.2	471	82
119,040	85.50	69.3	80.0	5.15	85.5	29.07	27.87	1.20	0.037	21,100	80.8	440	67
82,835	81.00	69.9	77.2	3.80	81.0	29.15	28.65	1.04	0.040	22,000	130	438	48
49,150	76.55	69.4	73.9	2.65	76.0	29.14	28.25	0.89	0.048	20,800	211	383	38

made by Mr. Bancel are with antiquated designs and disregard improvements of today. The aim of all condenser designers is to arrange the tubes composing the cooling surface so that they will be equally effective.

It is my opinion that the real efficiency of a surface condenser is measured by the terminal difference between the steam and circulating-water discharge temperatures, combined with a low-pressure drop through the condenser and hot condensate. Any steam pressure, or vacuum, has a corresponding temperature. If this temperature were the same as that of the circulating-water discharge, then the circulating water would have absorbed all the heat possible and the condenser, as a unit, would have an efficiency of 100 per cent.

To obtain low terminal difference requires: (1) Efficient tube layout so that all tubes are equally effective; (2) air box and baffle design to eliminate all stagnant air pockets.

That the aim of designers has been obtained is evidenced by the results in Table II from an official test made within the last few weeks. These results were obtained with a two-pass condenser containing 27,000 sq.ft. of cooling surface with the tube layout shown in the accompanying illustration.

It is interesting to compare a design such as recommended by Mr. Bancel with an improved two-pass design by data taken from the proposals at a recent offering for a large power plant. See Table III.

TABLE III—SINGLE VS. DOUBLE-PASS CONDENSER

	One-Pass Design	Two-Pass Design
Pounds of steam per hour to be condensed	175,000	175,000
Square feet cooling surface	14,250	27,000
Number of tubes	3,620	5,160
Size of tubes, in., 20 ft. long	1	1
Gallons per minute	36,000	36,000
Horsepower for pumps	110	114
Vacuum, in. mercury at 55 deg.	29.10	29.24
Heat transfer, B.t.u.	734	440

This offering would seem to disprove the claim of less pumping costs and in addition show that the two-pass design is much more conservative, offering a better vacuum, a greater factor of safety in heat transfer, surface, and what is more important, less outage on account of less frequent cleaning being necessary. And all at less first cost in dollars. FRANK R. WHEELER.

Chicago, Ill.

Effect of Hydrogen on CO_2

I was interested in the comments by B. M. Thornton, in the July 15 issue, in which he gave what he called an empirical rule for determining the excess air from the CO_2 percentage. In developing his equations, he multiplied by the ratio of the heating values of hydrogen and carbon. I cannot see how this ratio has anything to do with the relation between CO_2 percentage and excess air, but will show how simple equations can be developed that give the excess air accurately from the CO_2 percentage.

Let R equal the ratio weight of available hydrogen to carbon in the fuel; that is, $R = \frac{H - \frac{O}{8}}{C}$. Assuming that one volume of air is required to burn one pound of carbon, the resulting CO_2 will be 20.9 per cent of the volume supplied to burn the carbon. But for each pound of carbon air must be supplied to burn R pounds of hydrogen. Since three times as much air is required to burn a pound of hydrogen as is required for a pound of carbon, the quantity of air supplied for burning the hydrogen will be $3R$ times the volume supplied for burning the pound of carbon.

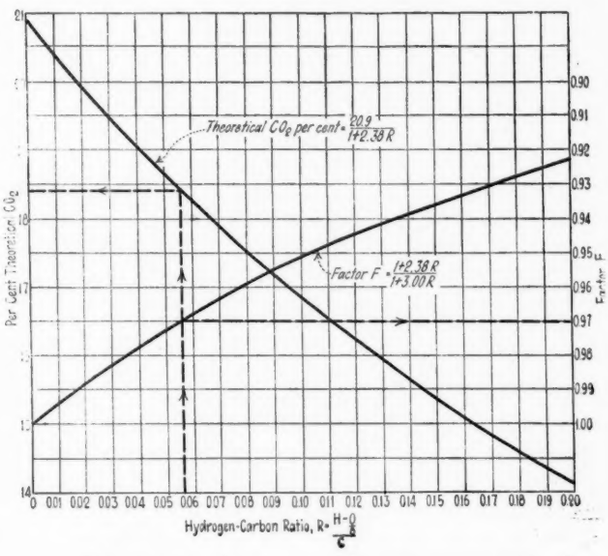
The total volume of air supplied per pound of carbon will be $1 + 3R$.

Of the air supplied for burning the hydrogen, the nitrogen alone will remain, the oxygen having united with the hydrogen to form water, which does not appear in the volumetric flue-gas analysis. With complete combustion—that is when the excess air is zero—the per cent of CO_2 will be $\frac{20.9}{1 + 0.791 \times 3R}$

When excess air is present, the per cent of CO_2 will be as follows:

$$CO_2 = \frac{20.9}{1 + 0.791\ (3R) + E\ (1 + 3R)}$$

in which E is the air supplied (expressed as a fraction



Effect of hydrogen on CO_2 and excess air

of the theoretical) in addition to what is theoretically required for combustion.

From the foregoing equation the following can be found:

$$E = \left(\frac{T}{CO_2} - 1\right) \frac{1 + 2.38R}{1 + 3R}$$

in which T is the theoretical CO_2 as given in one of the previously derived equations.

The accompanying curves show the values of the theoretical CO_2 and of the quantity $\left(\frac{1 + 2.38R}{1 + 3R}\right)$ for various values of the available hydrogen to carbon ratio. It will be seen that for a hydrogen to carbon ratio of 0.0556 the equation for excess air would be:

$$E = \left(\frac{18.5}{CO_2} - 1\right) 0.971$$

$$E = \frac{18.0}{CO_2} - 0.971$$

$$N = \frac{18.0}{CO_2} + 0.03$$

in which N is the ratio of the actual to the theoretical air supplied. For the same values for hydrogen-to-carbon ratio Mr. Thornton shows the equation to be $N = \frac{17.3}{CO_2}$. His equation would give too low a value for excess air. E. G. SUTTON.

Springdale, Pa.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Advantage of Swing Check Valves

What is the advantage of a swing check valve over the poppet style of check valve for use on a boiler-feed line? *N. A.*

Swing check valves are more nearly straightway in their passages than poppet check valves and are less likely to become stuck fast from wear or clogging of the guide for the valve, and as swing check valves seat more quietly, the valves and seats are subjected to less wear from pounding.

Additional Steam Required by Increase of Back Pressure

When the m.e.p. of an engine is 46 lb. per sq.in., how will the economy be affected by increasing the back pressure from 2 lb. to 4 lb. per sq.in.? *H. D.*

Development of the same power would require the same mean effective pressure, and an increase of 2 lb. per sq.in. of the average back pressure would require substantially as much steam as necessary to develop 46 + 2 = 48 lb. m.e.p. with the present 2 lb. back pressure. Under ordinary conditions this increase of load, amounting to about 2 per cent, or an increase of 2 lb. of the back pressure, would require the engine to be supplied with practically 2 per cent more steam.

Bulge in Ligaments of Tube Sheet

What would cause some of the ligaments between the tubes in the rear tube sheet of a horizontal return-tubular boiler to be slightly bulged out, and how can the irregularities be corrected? *S. E.*

If the bulging was not in the tube sheet before the tubes were set, it is likely that the buckling occurred in the process of expanding the tubes. If the tube ends remain tight, it is a fair indication that there is no dangerous movement of the heads and they may be regarded as reasonably safe. Upon removing the tubes, it probably would be found that much if not all of the deformation of the tube sheet remains as a permanent set in the material. In that case, if the bulge is to be removed, it should not be done by forcing or cold hammering, but by heating and "bucked-up" swaging.

Advantages of Radiators Over Long Wall Coils

For heating a large public hall, will warming by heating surfaces of radiators or by long wall coils require less fuel? *R. M.*

The same gross amount of heat will be obtained for the same expenditure of fuel. However, that system will be more economical which provides the distribution of heat that is best adapted to the requirements. Over-heating some parts of the building to accomplish sufficient heat of the remainder results in waste of heat and discomfort of the occupants. Under these conditions operation of a heating apparatus requires more coal unless the poorly heated places are neglected. Better distribution and regulation of the heating are obtainable by employment of a number of radiators than by long wall coils, and the saving of heat should result in a saving of fuel.

Allowable Pressure for Outside Sheet of Stayed Water Leg

How is the maximum allowable working pressure determined for the outside sheet of a stayed water leg of a vertical fire-tube boiler? *R. N.*

The maximum allowable working pressure must be computed without allowing for the holding power of the stays, due allowance being made for the weakening effect of the holes for the stays; and to this pressure there should be added the pressure for braced and stayed surfaces secured by the formula,

$$P = 100 \times \frac{T^2}{p^2}$$

where

P = Maximum allowable working pressure, pounds per sq.in.;
T = Thickness of plate in sixteenths of an inch;
p = Greatest pitch of staybolts;

provided, however, that the additional pressure secured by the staybolts by the formula shall not be taken as any greater than the pressure corresponding to the allowable stress for the stays, when each stay is assumed to resist the pressure acting on the full area of external surface supported by the stay.

Removal of Water from Ammonia Refrigeration System

From blowing out of a gasket on an old type of ice machine, we are troubled with considerable water in the ammonia. Could not the water be removed by blowing out the trap on the brine cooler, or is a regenerator necessary for separating the water, and if so what is the correct temperature? *N. E.*

For removal of the water with economy of ammonia it is necessary to employ a regenerator or purifier. The water cannot be eliminated by blowing of the trap at the brine cooler, as ammonia is highly soluble in water, since water will absorb 400 to 600 times its own volume of ammonia vapor. After water has circulated in the system a short time, it becomes aqua ammonia, and if blown out at the trap, there is loss of a large volume of absorbed ammonia vapor.

By properly heating the liquid in a regenerator, the

ammonia is driven off while the water remains, to be discharged separately. It is impossible to eliminate all the water, because in driving off the ammonia, a small amount of water vapor is also driven off. The action of the regenerator is the same as that of a generator in an absorption system. The correct temperature will depend on the pressure carried in the regenerator, which, in most installations, is the pressure on the suction side of the system.

Books on refrigeration quote a table which gives the correct temperature and corresponding steam pressure for any generator pressure required in an absorption system, and these pressures hold for the regenerator or reclaimer. If the regenerator is connected in the discharge side of the system, it will be necessary to have the temperature of the steam coils high enough to cause the ammonia to gas-off at the higher pressure.

Hard and Soft Boiler Patches

What is the difference between a hard and a soft patch for a boiler? *A. M. H.*

A hard patch consists of a piece of steel plate accurately shaped, riveted and calked as thoroughly as would be done in new construction of a boiler. Soft patches usually consist of soft wrought iron shaped to fit the required place by hammering cold. A soft patch cannot be calked. To prevent leakage, the patch is set in a putty of white and red lead with iron borings and coarse iron filings or other boilermaker's putty, or sometimes the joint is made with a lead or rubber gasket. The patch usually is held in place by patch bolts which consist of setscrews countersunk under the head for fitting countersunk holes in the patch when the patch bolts are drawn up tight in tapped holes previously prepared in the boiler sheet. For setting the patch bolts up tight, they are provided with small projecting heads. When they would be exposed to the heat of the fire, these heads are chipped off after the bolts are set, leaving a finish like the end of a rivet with a countersunk head. A soft patch sometimes is applied temporarily but never should be regarded as a permanent repair.

Loss from Use of Vented Receiver

When high-pressure heating returns at the temperature of 280 deg. F. are discharged into a receiver, with vent pipe open to the atmosphere, before the returns are fed to the boiler, what percentage more of fuel is required in generation of the steam at the pressure of 80 lb. gage? *C. B. H.*

The heat contained by one pound of steam at the pressure of 80 lb. gage, or 95 lb. per sq.in. absolute, is 1,185.4 B.t.u. above 32 deg. F., and if the condensate were returned to the boiler at the temperature of 280 deg. F., each pound of the condensate for reconversion into steam would have to receive 1,185.4 + 32 — 280 = 937.4 B.t.u. When the condensate is discharged to a vented receiver, the temperature is reduced to 212 deg. F. and usually there is loss of about 5 per cent of the water in the form of vapor escaping to the atmosphere. So that in place of returning the condensate to the boiler at the temperature of 280 deg. F., there is about 95 per cent returned at the temperature of 212 with a loss of 280 — (212 × 0.95) = 78.6 B.t.u. per pound of original condensate requiring, for evaporation into steam 78.6 × 100 ÷ 937.4 = about 8 per cent more fuel.

Correction of Bourdon Pressure Gage

Is there a screw adjustment on a Bourdon pressure gage for regulation of the gage in case it is too fast or too slow, or if not, how is a pressure gage corrected? *J. B.*

When the indications of a Bourdon spring pressure gage are wrong at any part of the scale, the correct indications cannot be obtained at more than one point by simply adjusting the relative position of the spring to the scale, unless the divisions of the dial chance to be adapted to the new conditions. Usually, the movement of the pointer is not uniform for equal increments of pressure. After derangement from overpressure or other cause, the readings may show the same general characteristics, but would not be corrected by a simple screw adjustment or movement of the pointer to a different position with respect to the gage mechanism. Correction of a gage generally requires a new scale or retraining the Bourdon spring to the old scale by one who is experienced in such work.

Location of Fusible Plug in Return-Tubular Boiler

Where is a fusible plug placed in a return-tubular boiler, why is it so located and what care should a fusible plug receive? *T. L.*

A fusible plug in a return-tubular boiler is placed in the rear head, not less than two inches above the upper row of tubes, measured from the line of the upper surface of tubes to the center of the plug. This location is taken because, with an ordinary return-tubular boiler setting, the rear head is exposed to the hottest gases that come in contact with a part of the boiler where the fusible metal would melt from becoming uncovered by water and give warning by escape of steam and quick qenching of the fire, before it would be likely that the highest row of tubes would become overheated from further lowering of the boiler-water level. To keep fusible plugs in efficient condition, their surfaces, both on the fire and water sides, must be often scraped clean.

Advantages of Double Eccentric Corliss Valve Gear

For a Corliss engine what are the advantages of a double-eccentric over a single-eccentric valve gear? *L. M. W.*

With a single-eccentric valve gear the load is limited to that which can be carried with cutoff at less than one-half stroke with given point of release and compression, and release and compression are obtained at the expense of reducing the range of cutoff. With a double-eccentric valve gear the load may be increased to require cutoff as late as about seven-eighths of the stroke, and as the steam and exhaust valves are operated by separate eccentrics, the range of cutoff by the steam valves is independent of the point of release and compression obtained from the setting of the exhaust valves. For the same release and compression, with the load within the range of cutoff of the single-eccentric engine, the steam required per horsepower-hour is the same whether there is a single- or a double-eccentric valve gear.

[Correspondents sending us inquiries, should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Here and There in the Power Plant

Sidelights on things generally used but less generally understood

Storing Steam in Hot Water

WHEN water at a temperature above 212 deg. is discharged to the atmosphere, part of it flashes into steam and the temperature falls at once to 212.

The fraction of the total weight thus evaporating depends only upon the initial temperature of the water. Roughly, one per cent of the water evaporates for each ten degrees that the initial temperature is elevated above 212. Water gives up practically 1 British thermal unit for each degree it falls in temperature. Falling 10 deg., it gives up about 10 B.t.u. At atmospheric pressure a little less than one thousand B.t.u. are required to evaporate one pound of water, so the 10 B.t.u. lost by the water is sufficient to evaporate 0.01 lb., or one per cent of the water. The *actual* amount evaporated would be as shown in the accompanying table.

STEAM PRODUCED BY RELEASE OF HOT WATER

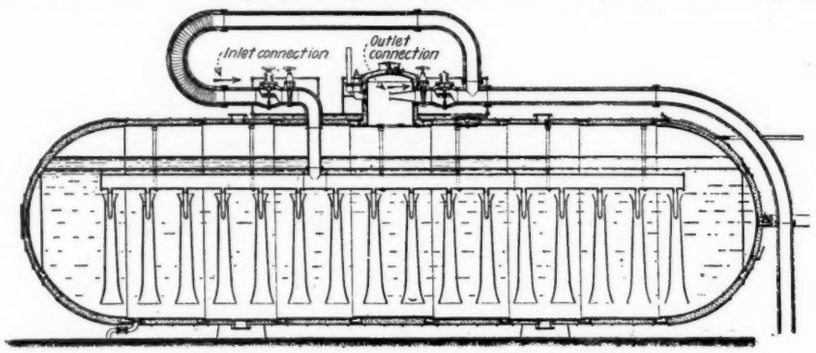
Initial Temp. of Water, Deg. F.	Saturation Pressure, Lb. Gage	Amount Flashing to Steam, Per Cent
212	0	0
220	2.5	0.8
230	6.1	1.9
240	10.3	2.9
250	15.1	4.0
260	20.7	5.0
270	27	6.1
280	34	7.1
290	43	8.2
300	52	9.2
320	75	11.3
340	103	13.5
360	138	15.6
380	181	17.7
400	232	20.0
450	406	25.5
500	669	31.3

The items in this table are easily figured. Take 300 deg. for example. The heat in the liquid at this temperature is 269.6 B.t.u. That at 212 deg. is 180 B.t.u., leaving an excess of 89.6 B.t.u. to evaporate water. At 212 degrees the latent heat is 970 B.t.u., so the excess heat is sufficient to evaporate $89.6 \div 970 = 0.092$ lb., or 9.2 per cent of the water.

This explains why a large storage of water constitutes such a source of danger in a boiler explosion. Fracture of any part of the boiler releases the pressure, which results in the production of enormous quantities of steam. The reaction of this jet of steam drives the boiler along much as a skyrocket is driven along by the discharge of explosion products.

The capacity of water to store large quantities of steam is made use of in the steam accumulator, which has already been discussed in the article "Storing Energy" in the previous issue. As there stated, the steam accumulator is a tank built to hold a large quantity of hot water under pressure and equipped with devices to assist the absorption of steam by the water when the pressure rises and its discharge when the pressure falls. The accompanying drawing shows such an accumulator in cross-section. This device serves as a flywheel for steam, storing it up when the demand is light and giving it off at the peaks.

Take a pulp mill, for example, where large amounts of process steam are required. The steam demand is very irregular, the sudden opening of a valve to a digester producing an enormous drain of steam for a few minutes while the liquor is heating up. If no means of energy storage is provided, enough boiler capacity must be installed to carry these peaks, after allowing for the limited heat-storage capacity of the boilers themselves. A steam accumulator connected into the system soaks up steam when the supply is greater than the demand and gives it off when the demand exceeds the capacity of the boilers. Thus less boiler capacity is needed. At the same time the boilers, operating at a practically steady load near the point of best efficiency, are more economical. A third advantage of the steam accumulator is a speeding up of production due to the fact that steam can be supplied as fast as needed during the peaks.



Steam accumulator showing tubes to assist absorption of steam

The steam accumulator is like a flywheel in another respect. In order to make practical use of the energy storage capacity of a flywheel, speed variations must be permitted. In the same way a steam accumulator cannot help in smoothing out the demand for steam on the boilers unless pressure variations are permitted. The larger the permissible variation in the pressure the greater is the amount of steam that can be stored.

The pressure on the accumulator varies through a wide range, say from 100 to 50 lb. gage. Starting with a period of low demand, the boilers are kept going at their normal rate. As a result the pressure gradually rises. This causes the steam to pass down through the nozzles of the accumulator and bubble up through the water. This steam is immediately condensed, raising the temperature of the water to the saturation temperature corresponding to the pressure. Assume that the pressure has been gradually raised from 50 to 100 lb. (gage). The temperature of the water in the accumulator will have risen at the same time from 298 deg. F. to 338 deg.

If now a sudden load comes on exceeding the normal capacity of the boilers, the pressure falls. Immediately the water in the accumulator starts to boil, adding its steam to that furnished by the boilers.

It might appear at first that the steam produced by an accumulator could be figured exactly by the method already used in figuring the flashing of water into steam when discharged to the atmosphere. There is, however, a fine distinction between the two cases. Where hot water is discharged to the atmosphere, all the steam is produced at atmospheric pressure or, at least, practically reaches that pressure before becoming disentangled from the water. The product then may be properly described as saturated steam at the final pressure.

Now take the case of a steam accumulator starting at 100 lb. gage. As the pressure gradually falls, steam is given off at a continually reducing pressure. Some of the steam is delivered at 100 lb. (practically), some at 90, some at 80, 70, 60 and 50. It is evident that the weight produced must be affected by this variation of pressure. However, within reasonable pressure limits, the error in assuming that all the steam is produced at the lower pressure is not serious.

On this basis the fall of pressure from 100 lb. gage to 50 lb. will reduce the heat of the liquid from 309 B.t.u. to 267.5 B.t.u., a fall of 41.5 B.t.u. The latent heat at 50 lb. is 911 B.t.u. The evaporation per pound of water would then be $41.5 \div 911 = 0.045$, or 4.5 per cent.

A Practical Test by Oxidation to Determine the Stability of Mineral Oil*

By T. S. SLIGH, Jr.†

SLUDGING of turbine oils is primarily due to oxidation, being especially evident where the oil is churned or sprayed, and thus finely divided and exposed to air. Oxidation is augmented by the presence of catalyzers, such as rust, bronze, copper, etc., as well as by high temperatures.

While no attempt has been made to link up the stability characteristics of an oil as obtained by testing with oxygen to the results of oxidation in the power plant, nevertheless this test appears to bear an important relation to the operation of lubricants in prime movers and other machinery.

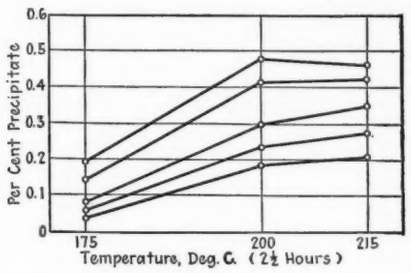


Fig. 1—High temperatures greatly increase oxidation in various oils

The test, briefly, consists of exposing a 10-gram sample of the oil in a closed flask of standard form and dimensions, which contains an atmosphere of oxygen. This is maintained at a temperature of 200 deg. C., or 392 deg. F. for 2½ hours. The result of oxidation is measured in terms of the precipitate or sludge which is thrown down when a certain portion of the exposed oil is placed in a definite amount of petroleum naphtha.

The oxidation number of the oil or, in other words, the measurement of its stability in the presence of oxygen, is the percentage of precipitate or sludge multiplied by one hundred. Thus, if the weight of the sludge after being filtered out was found to be one-half per cent of that of the sample exposed, the oxidation number would then be $0.5 \times 100 = 50$.

It is important that such tests of a given oil may be reproduced at any time with practically the same result. A study was made of conditions that might cause variation in results. Temperature must be held accurately to the required standard, while slight changes in the duration or time of the test are not of as great importance.

Since oxidation occurs in a closed vessel containing an atmosphere of oxygen, the effects of evaporation and the uncertainties of artificial ventilation are avoided. The temperature is regulated by placing the closed oxidation flask in a bath of stirred liquid which is held accurately at the prescribed temperature.

A remarkably uniform showing for this test on a diversity of lubricants was obtained. Apparently, the test could be repeated with a precision of about one-half per cent of the oxidation number. In other words, if a 10-gram sample of oil precipitated 0.51 per cent, making its oxidation number 51, the oxidation number of several tests would not be expected to vary more than from 50.75 to 51.25.

The majority of motor oils tested show oxidation numbers ranging from 20 to 75. Turbine oils range from 15 to 40, transformer oils from 5 to 20.

It was noted that in general, oils having a large oxidation number such as 70, did not behave so consistently as those having lower numbers. Turbine and transformer oils, therefore, belong in the consistent class. Traces of suspended matter increase the factor of variation in the tests.

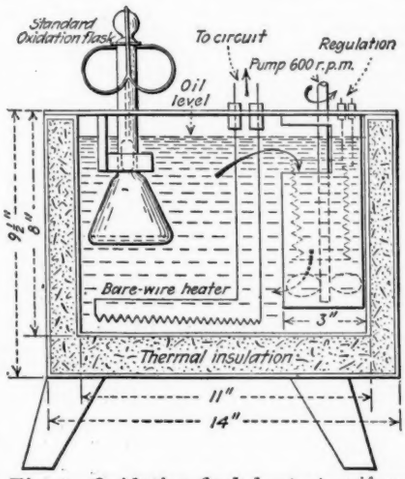


Fig. 2—Oxidation flask kept at uniform temperature by circulating bath

Oxidation of oil is accompanied by a discoloration, greater viscosity, increase in acidity, and the development of carbon dioxide and water, together with formation of precipitates of an asphaltic nature. There are three principal stages of oxidation, the first being the development of *asphaltic resins*, such as are ordinarily present in crudes. These are soluble in oil and in petroleum naphtha solution, being responsible for the darker color and increase of viscosity.

Second, formation of *asphaltene*, by the oxidation of resins in the first group, and which may be precipitated by dissolving the oxidized oil in petroleum naphtha.

Third, *varnish*, which is apparently the final oxidation product short of actual combustion and forms a strongly adherent film which sticks to the walls of vessels in which oxidation takes place. It is present to some extent in used oils. Varnish is characterized by its resistance to the action of ordinary solvents. Pyridine alone is reported as able to dissolve it.

The oxidation method of testing utilizes a temperature and period of time so chosen that the third group is ordinarily not formed, the oxidation number being determined by the percentage of asphaltene formed and precipitated out.

A general plan of the apparatus is indicated in Fig. 2, showing the standard oxidation flask manufactured especially for such purposes with a volume of about 100 c.c. This is kept at a constant temperature in an oil bath, the latter circulated by means of a pump and heated by an electric unit.

The precipitate is obtained by subjecting a definite portion of the oxidized oil to the action of naphtha and then putting this through a precipitating and filtering process involving specified material and routine. If desired to make this test accurately, a copy of the specifications should be obtained from the American Society of Testing Materials. Since there are only two weighing operations and no involved chemical procedure, there should be no difficulty in performing this test by anyone giving it the proper application and attention.

Figure 1 shows the effect of temperature during a 2½-hour test. For accurate determination this should be held as closely as possible to the specified amount. Varnish readily forms and oil vapors oxidize above 225 deg. C. From the curves in Fig. 3 it is evident that the actual time is not difficult to control for the required accuracy of testing. The dotted lines show results when the flask is filled with fresh oxygen after 2½ hours. Oxidation in an atmosphere of air yields only about 3 per cent of the amount precipitated with an equal exposure to pure oxygen. Carbon dioxide has little effect.

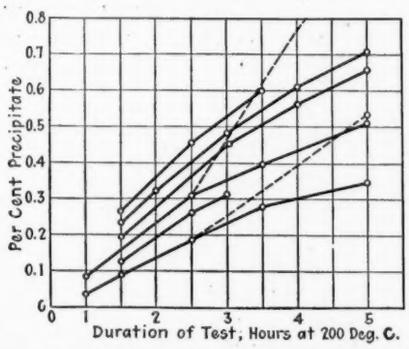


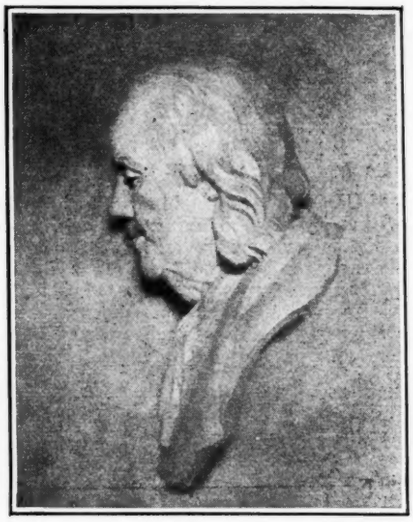
Fig. 3—How precipitate in different oils varies with testing time

*Abstract of paper, "An Oxidation Method for Determining the Stability of Mineral Oils," presented at the 27th annual meeting of the American Society for Testing Materials, and published by permission of the Bureau of Standards, United States Department of Commerce.

†Physicist, United States Bureau of Standards.

Franklin Institute Celebrates Its Centenary of Founding

Distinguished Scientists and Engineers from Europe, Canada and the United States Assist in the Celebration—Bartol Research Foundation Holds Inauguration



THE FRANKLIN INSTITUTE, which celebrated its century birthday in a three-day meeting, September 17-19, with an attendance of distinguished scientists and engineers from Europe, Canada and all sections of the United States, came into being through the efforts of a young man not yet 21, Samuel V. Merrick, who in 1823 thought that there should be an American institution similar to the institute established in London by Count Rumford.

His first effort at trying to interest scientific circles in this idea was unsuccessful, but dauntless and aided by Prof. William M. Keating, of the University of Pennsylvania, who had just returned from studying in Europe, a meeting of citizens was called by them, to organize "The Franklin Institute of Philadelphia," honoring Benjamin Franklin, the great Philadelphian, who had been dead thirty-four years. Its stated object was "the promotion of the useful arts in Philadelphia by extending a knowledge of mechanical science to its members and others at a cheap rate." At that time Philadelphia led New York in population by 5,000 in a total of 121,000.

That meeting was held on February 5, 1824, in the old County Court House and was well attended. A charter was drawn up soon afterward, and James Ronaldson was elected president. Lecture courses were established and a drawing school started, and the first exhibition of American industry was held at Carpenter's Hall. These exhibitions, predecessors of Philadelphia's famous exhibitions, became annual affairs and were of vital importance in the development of American industries. The exhibitions, notably those with which it celebrated its 50th anniversary and the still larger Electrical Exhibition of 1884 are also to its credit.

Program of Speeches and Papers Given at the Celebration

Academic Procession.

Address of Welcome by Hon. W. Freeland Kendrick.

Address by Wm. C. L. Eglin.

Address by Prof. Elihu Thomson.

"The Natural and Artificial Disintegration of Elements," by Prof. Sir Ernest Rutherford, Trinity College, Cambridge, England.

Unveiling of Tablet at Bartol Research Foundation.

"The Fifth Estate," by Arthur D. Little.

"Stimulation of Research and Invention," by Prof. D. S. Jacobus.

"Recent Developments in Aëronautics," by Prof. Joseph S. Ames.

"Military Aircraft and Their Use in Warfare," by Maj. Gen. Mason M. Patrick.

"Electrical Communications for Military Purposes," by Maj. Gen. George Owen Squier.

"Progress in Telephony," by Gen. John J. Carty.

"Concentration and Polarization at the Cathode During Electrolysis of Solutions of Copper Salts," by Prof. W. Lash Miller, University of Toronto.

"The Theory of Color Reproduction in Organic and Inorganic Compounds," by Prof. Julius Stieglitz.

"Photo-Elasticity," by Prof. E. G. Coker, University College, London.

"The Progress and Promise of Engineering," Dexter S. Kimball.

"Some of the Effects of Loading Granular Material," by Daniel E. Moran.

"Modern Military Engineering," by Brig. Gen. Edgar Jadwin.

"The Way of the Wind," by Prof. W. J. Humphreys.

"Modern X-ray Tube Development," by William D. Coolidge.

"Inorganic Crystals," by Prof. William Lawrence Bragg, Victoria University.

"Technical Results of the Theoretical Development in Chemistry," by Prof. F. Haber, Direktor, Institut fur Physikalische Chemie and Elektrochemie.

"The Development of Colloid Chemistry," by Prof. Wilder D. Bancroft.

"Applied and Scientific Photography," by C. E. K. Mees.

"Some Aspects of High Pressure Research," by Prof. P. W. Bridgman.

"The Phonodeik," by Prof. Dayton C. Miller.

"Electronic Phenomena at the Surface of Metals," by Prof. Charles E. Mendenhall.

"The Carbon Atom in Crystalline Structure," by Prof. Sir William Henry Bragg, Royal Institute.

"Some Causes of Volcanic Activity," by Arthur L. Day.

"Spectroscopy in Past and Present,"by Prof. Charles Fabry, University of Paris.

"Infra Red Spectroscopy," by Prof. Augustus Trowbridge.

"Lightning," F. W. Peek.

"The Measurement of Acoustic Impedance by the Aid of the Telephone Receiver," by Prof. A. E. Kennelly.

"The Field of Research in Industrial Institutions," by E. W. Rice, Jr.

"Unusual Problems Encountered in the Design and Construction of Large Bridges," by Ralph Modjeski.

"Steam Turbines on Land and Sea," by Sir Charles Algernon Parsons, Newcastle-on-Tyne.

"Mercury Boiler," by William Leroy Emmet.

"The Origin of the Earth's Electric and Magnetic Phenomena," by Prof. W. F. G. Swann.

"Radiating Atoms in Magnetic Fields," by Prof. Pieter Zeeman, University of Amsterdam.

"Motion of Electrons in Gases," by Prof. John Sealy Edward Townsend, University of Oxford.

"Electric Discharges Between High Resistance Electrodes," by J. Slepian.

"Twenty-Five Years' Progress in Explosives," by Charles L. Reese.

"The Influence of J. Willard Gibbs on the Science of Physical Chemistry," by Prof. F. G. Donnan, University College, London.

"Early Science in Philadelphia," by Provost Emeritus Edgar F. Smith.

"The Trend in Physical Metallurgy," by Prof. C. H. Mathewson.

"The Restraint of Exaggerated Grain Growth in Critically Strained Metals," by George L. Kelly.

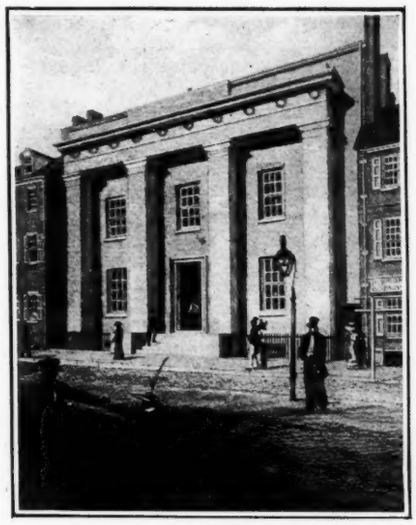
"Magnetic Analysis of Steel," by Prof. Bradley Stoughton.

"Magnetic Properties of Fifty Per Cent Iron Fifty Per Cent Nickel Alloys," by T. D. Yenson.

"A New Type of Non-inductive High Resistance," by Dean Harold Pender.

"Results of a Preliminary Determination of the Velocity of Light," by Prof. A. A. Michelson.

"Modern Ordnance," by Maj. Gen. C. C. Williams.



Franklin Institute, Erected in 1825

Its journal was founded in 1826 and is still published regularly. Its museum contains many valuable and interesting exhibits, including Franklin's electrical machine, a picture of which is shown in this article.

Its committees have conducted investigations and researches upon many subjects such as the efficiency of waterwheels, the prevention of fires in theaters, weights and measures, and it has initiated and fostered wise projects of legislation, national as well as local.

On June 8, 1825, "at high 12 o'clock, the cornerstone of the Hall of the Franklin Institute was laid in ancient and Masonic form by the Grand Lodge of Pennsylvania in the presence of the society." It contains a lecture room capable of seating some 300, from the platform of which many of the world's great scientists and engineers have presented the results of their work and have had these results recognized through the awards of the medals of the organizations. These awards are: The Franklin Medal, the Elliott Cresson Medal, the Howard N. Potts Medal, the Edward Longstreth Medal, and a Certificate of Merit. Among the recipients of the medals have been Thomas A. Edison, Guglielmo Marconi, Professor Michaelson, Sir Charles Parsons, Sir Joseph Thomson, Frank J. Sprague, Ralph Modjeski, General John V. Carty, Alfred Noble, Sir Robert A. Hadfield, Menri Moissan, Dr. Michael I. Pupin, Professor Roentgen, Elihu Thomson.

Engineers have always played an important part in the management of the Institute. Among the presidents have been Coleman Sellers, William Sellers and John Birkinbine. The president at the present time is also an engineer, Dr. William C. L. Eglin, vice-president and chief engineer of the Philadelphia Electric Co., whose picture accompanies this article.

The Celebration

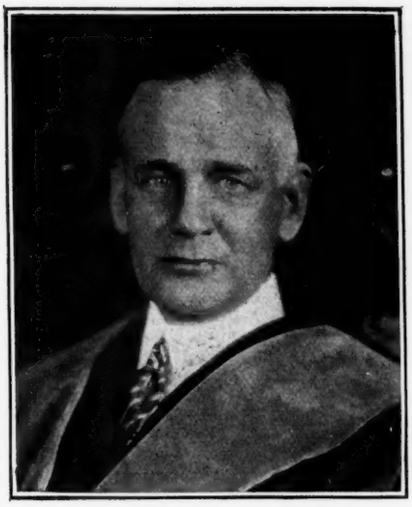
Nearly fifty papers on engineering and scientific subjects were presented at the group sessions, and the entire attendance was brought together in three general meetings.

The gathering was attended by some 800 distinguished men and women, some of whom presented papers and many of them being the designated representatives of universities and scientific societies in every part of the world. The opening session was held at the Walnut Street Theater, with a large audience—many of whom were in academic costume. The sectional meetings were held in Franklin Institute and various halls of the University of Pennsylvania.

Any attempt to summarize even briefly the proceedings as a whole would take a great deal of space, but a glance at the program which is included in this article will show the scope of the subjects discussed. On Friday afternoon the Honorary Degree of Doctor of Science was conferred with impressive ceremonies by the University of Pennsylvania upon Prof. Sir William H. Bragg; Prof. Charles Fabry; Sir Charles Algernon Parsons; Dr. E. W. Rice, Jr.; Prof. Pieter Zeeman; Dr. William C. L. Eglin.

The concluding ceremony of the celebration was a gala banquet at which the delegates from universities and learned societies presented their congratulations to the Franklin Institute, and at which Dr. Haber, of nitrogen-fixation fame, gave an inspiring speech in which he characterized American and European relationship and the importance of natural science:

"The history of a young country may be divided into three epochs. The first is the period of commerce. A sparse population gathers together the natural treasures of the soil and throws them as crude products upon the world market. They exchange these for the products which are the results of a higher development, which come from the hands of an older civilization. We see this state of affairs today in Spanish South America. This period was long past when I traveled in the United States twenty-two years ago.

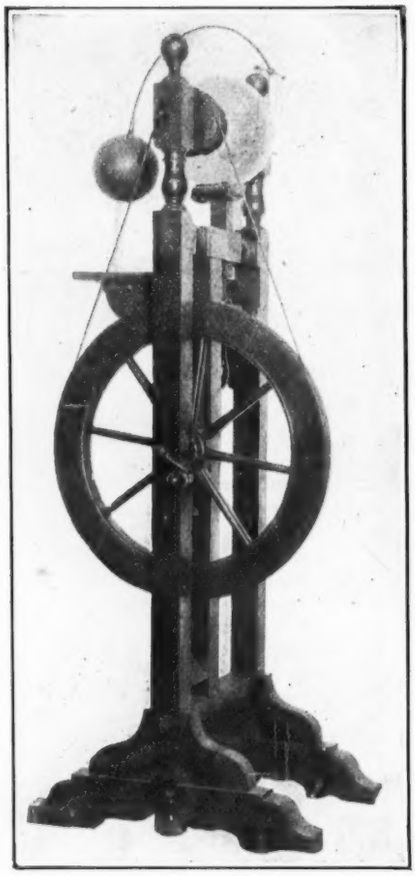


President Wm. C. L. Eglin

"When the population becomes denser and finer qualities in needs and deeds become evident we come to the period of technical development, in which the country takes over the processes from the people of older culture and applies them to its demands. Experience wanders from the old world and naturally takes a new form, often more powerful and more impressive than under the closer and narrower surroundings of the country in which it originated. The United States was twenty-two years ago in this state of technical progress. It was astonishing what you had then accomplished, but you were destined to still greater performances.

"Today your nation appears to a visitor like one of the countries of the old world culture. The first period of commerce lies far in the past; the second period of technical progress is also gone, and the third period, that of

Lecture Hall of Franklin Institute



Franklin's electric machine

Science, has begun, in which the country is not dependent upon the old world for either products or processes, but devises with its own creative genius the methods of work by whose aid it prepares its products.

"It is not now the congratulations of the wise teacher to the aspiring pupil that I extend. It is the respect of the older expert for the independence and splendid results accomplished by the younger colleague.

"In the last twenty years the world has undergone an immense upheaval. The lawyer, the soldier, the financier, and the man of industry have emulated the striving after world supremacy and have overthrown the order of things. Now they are disputing with each other about reconstruction, and, for the ordinary looker-on, it seems as if in place of the reigning princes, who for centuries governed the world we now have the controlling banker. But this is only a superficial aspect. The banker and lawyer, the industrialist and merchant, in spite of their leading positions in life, are only administrative officials, and the sovereign is Natural Science. Its progress determines the measure of the prosperity of man, its cultivation is the seed from which the welfare of future generations grows."

Sawmill Waste as a Fuel*

By Otto Nordström†

The world's production of sawn-wood goods probably amounts to fifty or sixty billion board feet per year. Of this amount about two-thirds comes from the United States. In Sweden more than four-billion board feet are produced, about half of which are exported. By cubical measure the milling waste including sawdust, edgings and mill ends, amounts to as much as the finished product. To this must be added the sawdust and shavings from the planing mills.

Today one may see, both in the United States and in Canada, refuse burners in use for burning the mill waste, while large steam central stations near-by are, at the same time, fired with coal. In Sweden practically all the sawmill waste is utilized.

To burn the waste directly and with good efficiency is very difficult. There has been great need for some practical way of drying it. The following method has been operating since May of last year at the sulphide pulp mill of J. W. Enqvist AB., Tammerfors, Finland.

The boiler house contains three 3,000-sq.ft. horizontal water-tube boilers, with economizers. The flue gases used for drying are drawn from the flue, after passing through the economizers, by an electrically driven fan. The waste fuel arrives at the mill in railway trucks and is discharged from these to a conveyor which transports it to the tower's elevator. After being dried, the fuel is conveyed into the boiler room, where it is distributed to the furnaces. The speed of discharge from the towers may be regulated.

The tower, having a capacity of about 2,500 cu.ft. of fuel, consists primarily of inner and outer round shells of sheet iron, between which the fuel sinks down during the drying process. Both the inner and outer towers are made sloping to prevent the chips from bridging.

The hot gases from the boilers are delivered through a fan, which forces them from below into the inner tower and from this through radial perforations, the layer of fuel and the perforated outer tower, to the gas-collecting chamber and chimney.

†Consulting engineer, Sundsvall, Sweden.
*Abstract of paper presented at World Power Conference, Wembley, England.

Franklin Institute's banquet to delegates and speakers

Distinguished Foreign Engineers Entertained at Luncheon

The presidents and governing boards of the four Founder Engineering Societies gave an informal luncheon at the Engineers' Club, New York, last Friday in honor of Sir Charles Morgan, President of the Institution of Civil Engineers, Sir Charles A. Parsons, Senatore Luigi Luiggi, President of the Italian Society of Engineers, and William H. Patchell, President of the Institution of Mechanical Engineers. Over eighty members and other outstanding engineers were present to meet the distinguished guests, who came to this country to attend the sessions of the recent Centenary Celebration of the Franklin Institute.

In introducing the speakers, John W. Lieb referred to the participation of engineers in public affairs in foreign countries, particularly Italy. He read messages of regret from Sir Charles Morgan and William H. Patchell, who were unable to be present. Brief addresses of welcome on behalf of their respective societies were made by C. L. Loweth for the A.S.C.E.; J. V. W. Reynders for the A.I.M.M.E.; Farley Osgood for the A.I.E.E., and F. R. Low for the A.S.M.E. Mr. Low referred briefly to the hospitality extended to the American engineers on their visit to the World Power Conference last summer and expressed the conviction that the Conference had accomplished much in bringing together engineers of all countries for an exchange of ideas.

Parsons Would Investigate Depths of Earth

In response to the addresses of welcome, Sir Charles Parsons referred to the enormous power development in this country, especially that produced by steam turbines, and remarked, incidentally, that inasmuch as the initial cost of steam power is only about half that of hydro-electric, we may look largely to further steam developments to meet the major part of our power needs for many years to come. He also touched upon his favorite theme, that in the interest of science an international movement among engineers be started to investigate the depths of the earth by sinking a well twelve miles deep. This depth he thought could easily be attained without encountering prohibitive temperatures.

Senatore Luiggi reviewed the work of the early engineers and was inclined to give greater credit to the modern engineer, whom he characterized as one that not only builds structures, but gives life to them by making them of service to mankind. With reference to the status of engineers in Italy, he told of the long fight that had been necessary before recognition had been obtained, and mentioned a recent law that has been passed definitely establishing the status of engineers and making it possible in the future for only technical graduates to be so classed, with the exception of a few in the present generation who have attained prominence without such training and whose credentials are passed upon by a special committee appointed for that purpose.

The speaker also referred to the policy of the Italian government in encouraging public works and public-utility extension under private initiative but with Government subsidy. This applies to the reclamation of marsh land for the purpose of growing wheat, to irrigation and to hydro-electric developments. The government subsidizes such projects in some cases up to 60 per cent of the cost. To encourage greater development of hydro-electric power, the government will pay 40 lire per horsepower per year for a period of fifty years.

Senatore Luiggi referred to the Apulia aqueduct, which is perhaps little known to American engineers but which incorporates some unusual engineering features. Counting all branches and the aqueduct proper, it would total nearly two thousand miles and includes three tunnels of 6, 7 and 8 miles respectively. It supplies a population of a million and a half and the fall in head at several points is utilized to generate electric power, which in turn is applied to pumping the water to communities above the aqueduct level. This project was carried through successfully under private initiative with the Government subsidizing up to 80 per cent of the cost.

In this connection Senatore Luiggi referred to the socialistic period through which Italy had recently passed and to the present policy of Signor Mussolini in encouraging private initiative even to the point of turning back to private operation many of the utilities that had been placed under State control. This policy has had a marked effect on eliminating unemployment in Italy, where out of a population of forty million there are at present only 125,000 unemployed, many of whom are not working because of malaria prevailing in certain sections.

Those present at the luncheon were: W. H. Aldridge; E. H. B. Anderson; Sidney H. Ball; H. H. Barnes, Jr.; L. Barzini; C. E. Beam; A. D. Blake; N. F. Brady; W. E. Bullock; Lincoln Bush; N. A. Carle; Dr. J. McKeen Cattell; G. H. Clevenger; Mr. Chasworth; C. E. Davies; J. Vipon Davies; A. De Diasi; J. V. N. Dorr, A. S. Dwight; E. Eby; W. W. Erwin; A. D. Flinn; K. G. Frank; Carleton Greene; W. F. M. Goss; John Hazen; J. P. Hogen; C. M. Holland; Otis E. Hovey; J. M. Howe; Clemens Herschel; John Price Jackson; W. B. Jackson; D. S. Jacobus; Sidney J. Jennings; Peter B. Junkersfeld; James F. Kemp; George F. Kunz; Dr. Koettgen; Henry A. Lardner; W. H. Lawrence; J. W. Lieb; Fred R. Low; C. F. Loweth; Luigi Luiggi; Mario Luigi; E. C. Mathewson; William McClellan; C. W. Mailloux; E. J. Mehren; Malcolm Muir; Thomas E. Murray; A. Marson; George A. Orrok; J. F. O'Rourk; N. Nelson; Sir Charles Parsons; Ray Palmer; G. H. Pegram; J. P. H. Perry; John Platt, Jr.; Mario Puritz; Charles F. Rand; J. V. V. Reynders; S. G. Rhodes; T. B. Rhodes; Calvin W. Rice; R. Sanford Riley; H. B. Seaman; C. H. Sharp; F. F. Sharpless, F. W. Smith; J. E. Spurr; F. A. Stratton; Ambrose Swasey; Elmer A. Sperry; Calvert Townley; Philip Torchio; Dr. Van der Pijl; Paul D. Vecci; J. A. Vanderpool; W. E. Wickenden; Arthur Williams; Roy Wright.

Phases of the St. Lawrence Situation

One of the difficulties reported in connection with the discussion relating to the St. Lawrence development is that the waterway, which is inseparable from the power project, would involve getting around four obstacles, three of which are on Canadian soil and one in the United States. Canada is desirous that the United States share the expense of the necessary canals, but is unwilling to share equally the power thus made available. Aside from this is the insistence of Canada that the Chicago diversion be made part of the whole problem and the desire of this country not to include this in the deliberations. There is also the political opposition from southern New York, which fears that injury would be done to shipments through the Port of New York.

As far as St. Lawrence power is concerned, central New York, along the route to the Erie Canal, would be the logical section for its utilization. Here industries might be built up which would not only avoid greater concentration in the large cities, but would also increase the freight along the Erie Canal to the benefit of the Port of New York which might ultimately gain as much as it would lose through diversion of traffic through the Great Lakes.

Rensselaer Celebrates Its Centennial

The Rensselaer Polytechnic Institute, Troy, N. Y., the oldest college of science and engineering in any English-speaking country, is to celebrate its centennial Oct. 3-4.

There will be present official delegates from educational institutions, scientific organizations and engineering societies throughout the world, as well as from each alumni association and from each class since 1857. The program of exercises is as follows:

PROGRAM

Friday Morning—Academic procession; convocation; addresses by Herbert Hoover, Secretary of Commerce; Frank Pierrepont Graves, president of the University of New York; Harry E. Clinton, mayor of Troy; Frank Pierrepont Graves, president of University of New York State; Sir Charles Langbridge Morgan, president of the Institute of Civil Engineers of Great Britain; Henri Abraham, president of the Society of Electrical Engineers of France; Luigi Luiggi, president of the Society of Civil Engineers of Italy; Arthur Surveyer, president of the Engineering Institute of Canada; Palmer Chamberlaine Ricketts, president of Rensselaer Polytechnic Institute.

Friday Afternoon—Unveiling of tablets: (1) On the site of the old Main Building, destroyed by fire in 1904; addresses by Hon. Seymour van Santvoord, of Troy, and Mrs. James Carroll Frazer, of Washington, a lineal descendant of Stephen Van Rensselaer, founder of the Institute. (2) In the Carnegie Building in memory of Samuel Wells Williams '32, first professor of Chinese and Oriental literature in Yale University; address by Dr. Sao-Ke Alfred Sze, Chinese minister to the United States; Inspection of Laboratories.

Friday Evening—Dinner in the state armory; addresses by Livingston Farrand, president of Cornell University; Joseph Henry Odell, Director of service Citizens of Delaware; Huger Wilkinson Jervey, Dean of the Law School, Columbia University; pageant on the campus illustrating scenes in the history of science and engineering during the last hundred years.

Saturday Morning—Convocation; addresses by President Angell of Yale University, President Birge of the University of Wisconsin, President Stratton of the Massachusetts Institute of Technology, President Michelson of the National Academy of Sciences, President Grunsky of the American Society of Civil Engineers, President Low of the American Society of Mechanical Engineers, President Kelly of the American Institute of Mining and Metallurgical Engineers, President Osgood of the American Institute of Electrical Engineers.

Saturday Afternoon — Luncheon in Sage Hall; Alumni meeting; Inspection of Buildings and Laboratories; Reception by President Ricketts and Mrs. Ricketts; alumni meeting.

Saturday Evening—Pageant; alumni smoker.

Ruhr Competition May Affect British Coal Practice

A degree of care is exercised in the British coal fields to prevent the breaking of coal that greatly impressed the American visitors who were in attendance at the World Power Conference. Such a degree of care is entirely precluded in this country by high labor cost. Similar care is exercised to keep coal clean. In South Wales, for instance, there is universal use of a sort of exaggregated dustpan for loading coal in mine cars. The mine owners admit that the use of this "box," as it is called, restricts the amount of coal the men can load, but they explain that the use of regular shovels would result in dirt being dug up from the bottom.

South Wales continues to pay by the ton of "large" coal. At the tipple the coal passes over 1¼-in. bars and is screened, weighed and recorded. One of the reasons why they use so little powder is the fact that they do not wish to break up the coal. Fortunately, most British coal comes down easily. The desire to prevent breakage has become inborn. The British regard our mechanical trimmer as all but a criminal device. They are in no way interested in installing this "Yankee notion." They are convinced that their reputation in international trade is built largely on size. With the release of the Ruhr many Americans will not be surprised to see the competitive situation become such as to force the abandonment by the British of some of these practices which add to the cost of their coal.

Another practice which astounds visitors from the United States is the separation of coal coming from the same seam when there is a difference in its physical character. In Great Britain there frequently is a stratification within the seam such as is rarely encountered in the United States. The famous Barnsley bed, in Yorkshire and the Midlands, for instance, carries an extremely hard steam coal in one part of the seam and a much favored house coal in another. As this coal is loaded the steam coal is put in one car and the house coal in another. The steam coal is dumped on one set of screens and the house coal on another. Pickers re-sort any lumps that may have been classified wrongly by the loaders underground. It is contended that the sorting adds very little to the cost of the coal, but does add very materially to the sales realization.

This rather common sharp difference of qualities in the seams has led to an interesting practice in sampling. We think of a coal seam as a homogeneous thing. Our approved method of sampling is to cut a channel through the seam from top to bottom and quarter the coal until the sample has reached a size that can be handled conveniently. In England, however, separate analyses are made for each layer composing the bed. They frequently reveal marked variations in the value of the coal. In one bed, for instance, this practice revealed that a coal that formerly was penalized because of its sulphur content was getting all its sulphur from a narrow band in one part of the seam, which could be picked out easily. With that band removed, the coal was marketable for metallurgical coke.

The Americans found the British immensely interested in low-temperature carbonization. They are striving to attain a smokeless fireplace fuel which can be burned readily in the type of grate found in practically every room of a British residence.

This work also is being encouraged by the demand for motor spirit. The British are anxious to render themselves less dependent on the outside world for motor fuel. Some of the low-temperature processes claim a larger yield of motor spirit than is obtainable as a result of byproduct methods. No less than six large semi-commercial plants are being operated on low-temperature processes. Another half dozen methods are in various stages of promotion, but have not reached the point of erecting an experimental plant.

In the opinion of the Americans who looked over the low-temperature situation a commercial success has been attained for no one of the processes. They do produce a fine coke which burns freely without smoke. Considerable amounts of tar and oils are recovered, but the recoveries are not sufficient to meet the capital charges, it is stated.

The London Coke Committee, a co-operative body on which all the gas companies are represented, have solved the problem of finding a market for their coke and breeze. A portion of it, however, is being used successfully under boilers.

British Imports and Exports of Machinery

British imports of machinery during the first six months of 1924 advanced slightly in comparison with the corresponding period in 1923, according to *Commerce Reports*. Imports of prime movers increased by 1,700 tons. Imports of electrical machinery decreased by 169 tons. The following tables give British imports for the first six months of 1923 and 1924:

BRITISH MACHINERY IMPORTS, JANUARY TO JUNE, 1923 AND 1924

Class	January to June, 1923 Tons	January to June, 1923 Value	January to June, 1924 Tons	January to June, 1924 Value
Boiler and boiler house plant	373	$170,685	435	$237,779
Electrical	2,293	2,407,912	2,124	2,317,223
Prime movers	1,773	822,872	3,475	1,567,518

Exports of prime movers, January to June, showed an increase during the later months.

The following table gives British exports of machinery for the first six months of 1923 and 1924:

BRITISH MACHINERY EXPORTS, JANUARY TO JUNE, 1923 AND 1924

Class	January to June 1923 Tons	January to June 1923 Value	January to June, 1924 Tons	January to June, 1924 Value
Boiler and boiler house plant	27,011	$6,289,950	26,669	$5,636,717
Electrical	9,887	8,449,496	15,210	10,856,016
Prime movers	12,423	5,040,788	14,896	6,080,351

Memphis Company To Install New Unit

The Memphis Light & Power Co., which supplies light and power to Memphis, Tenn., has, according to recent press reports, just completed the first unit on an addition to its lighting plant, the expenditure for which approximated $3,000,000. Work is being pushed on the second unit, the decision being reached to install this a year in advance of the plans as originally determined in making appropriations for the improvement.

Two Large Power Systems To Be Linked

The Niagara, Lockport & Ontario Power Co. and the Pennsylvania Electric Corp. announce that these two large systems will interconnect their 110,000-volt transmission facilities at the New York-Pennsylvania State line. The purpose of the linking up of the two systems is to avoid duplication of investment and realize increased efficiency and economy in generation and distribution of electric energy. The system will unite the water-power development of Canada, New York and the New England States with the steam-power plants in Pennsylvania, and also hydro-electric developments in western Maryland and Pennsylvania.

Old Deep River Coal Mines To Be Utilized

As the result of an increased demand for electricity by the industrial plants of North Carolina, the authorities of the state are contemplating the early working of the coal deposits in the Deep River district. These deposits have been regarded as commercially unprofitable, but with the present output of hydro-electric energy insufficient for the state's needs, even when supplemented by a large output of power generated by steam, the Deep River mines are regarded as the only additional source of the needed power, according to press reports.

The excess demand for electrical energy over the production within the State is now supplied by power transmitted from South Carolina and other neighboring states whose industrial development has not yet reached the point of utilizing all of their hydro-production. However, according to the North Carolina Geological Survey, it is only a question of time before this situation will be altered and North Carolina will be deprived of this source of power. By far the greater part of the State's water-powers have been developed and sources of supply are sharply limited, and while there will be great and more intensive development of water power, still more and more coal must be used in steam plants not only to supplement and stabilize the output of hydro power, but in time to produce electrical energy independently.

In a time when the steam plant is being developed to get the utmost power out of every ton of coal consumed and when the use of powdered coal is a proved success, it is a reasonable vision to see the Deep River field producing 1,000,000 tons a year, instead of the approximately 100,000 tons which is the capacity of the present mines. The Deep River field has abundance of water for condensing purposes.

N. Y. Water-Power Commission Fixes Niagara Power Rental

The New York State Waterpower Commission announced on Sept. 23, the fixing of a rental of 50 cents per horsepower for power derived from Niagara Falls in excess of 15,100 cu.ft. per sec. by the Niagara Falls Power Co. It was stipulated that this rental would be operative only on the condition that the company itself shall pay the rental and shall not pass it on to the distributors.

At the hearing on the problem held at Albany, Sept. 22, representatives of the Niagara Falls Power Co. said the law requiring fixing of such rentals was not mandatory and that the enforcement of the law would only be passed on to the ultimate consumer. The matter of overdue state rentals from the St. Lawrence River Power Co. at Massena fixed by the commission at $19,125 a year in 1921, payment of which has not as yet been made by the company was also taken up, but action was postponed until Oct. 15, when the officials of the company, who are now in Europe, would be able to be present.

Representatives of the Niagara Power Co. said they were without power to accept or protest the commission's ruling, but would lay the question before the company's board of directors. They said the commission would be informed of the company's decision within a few days.

The commission denied the application of several companies for the private development of water power aggregating about 4,300,000 hp. on the rivers of the State. Included in these applications were those of Lower Niagara River Power Co., the Niagara Gorge Power Co. and the Moose River Power Co.

Austria Forging Ahead with Her Power Schemes

Dr. Rudolf Reich, chief of the electrification department of the Ministry of Commerce and Traffic, says in an article in the *Christian Science Monitor* of recent date:

"Just a month ago the main gallery of the Ylls Hydraulic Power station at Opponitz was pierced after two years' hard work. This brings the first of the larger power plants which are to supply Vienna with electricity near completion. It is hoped that it will be ready for use this autumn. The city was badly off for coal when the overthrow of 1918 cut Austria from her coal resources, since Vienna's coal consumption amounts to almost a third of that of the entire country. The dire need of coal drove home the realization all the sooner, however, of the necessity of exploiting the sources of power in this country.

"Vienna, it is believed, will be provided within a year with all the power required. At present the city consumes 300,000,000 kw.-hr. Of this amount, the aforementioned Opponitz works are to supply 56,000,000 kw.-hr., but it is anticipated that the remaining 244,000,000 kw.-hr. will be taken care of by two more power stations, the building of which are now under contemplation."

Dr. Reich believes that Austria has sufficient water power to cover all her own needs, and of this at present only 13 per cent has been turned into use, with an additional 4 per cent under construction. The employment of this water power is "to reduce to a minimum the consumption of coal."

Austria is taking much interest in the electrification of her railways, which is one of the main incentives to water power development.

Power Show's Field Expands This Year

The Third National Exposition of Power and Mechanical Engineering will be held in the Grand Central Palace, New York, Dec. 1-6, 1924. The preceding shows have thoroughly covered the field of power production with the exception of hydraulic machinery, but in the coming event there will be an excellent representation of the manufacturers of large turbines and hydraulic power-plant equipment.

The large increase in the use of the individual motor drive on machine tools has led the managers of the coming show to make a determined effort to bring a group of manufacturers of modern economical motor-driven machine tools. The field of power transmission and use has been one of great interest to visitors to the Power Show, and the exhibits of bearings, couplings, belting and belt drives and gearing have always attracted large groups. They will be extended and elaborated at the coming exhibit.

High pressures and temperatures brought into the power plant for increasing economy of fuel consumption have also brought added complications, and the majority of these are in the piping and fittings. Power-plant engineers will therefore look with great interest for the exhibits in this line, in which many advances have been made during the year, of the many manufacturers of valve and fittings represented.

The showing of different types of fuels will be elaborated and extended this year. Large scale models of modern power plants will also be displayed. The lectures to be given at the Power Show will be arranged so that they will not conflict with the sessions of the American Society of Mechanical Engineers and the American Society of Refrigerating Engineers, whose annual meetings will be held the same week.

Experiments have been made in the furnaces at Cahokia (St. Louis) on burning pulverized petroleum coke. This material is a byproduct of the oil-refining industry and has hitherto been considered a waste. Should it prove to be successful, it will furnish a desirable addition to the regular coal supply. Belleville bituminous coal, which is unsatisfactory as a stoker-fired fuel is being used in the pulverized form quite well.—*Wire and Pipe.*

Kansas City, Kan., Light & Power Plant Reports

The Kansas City, Kan., Department of Water, Light and Power, L. H. Chapman, Commissioner, has recently issued its 1923 yearbook. The report shows that the gross income from operations was $1,197,397.89, of which prime operating costs were $944,087.45, or 79 per cent. The net income before fixed charges was $268,437.39, or 22½ per cent.

The equipment of the generating plant consists of turbo-generators of 23,750 kva. capacity and boilers of 62,040 sq.ft. of heating surface.

The maintenance expenditure was: $27,922.98 on boiler room equipment; $10,133.99 on stokers and equipment; $4,088.80 on turbines and equipment; $4,535.77 on generators and equipment; $2,763.27 on conveyors and equipment; $3,284.58 on condenser equipment; $1,079.19 on turbine fan and equipment. The following table gives the cost of operation:

COST OF GENERATING ELECTRIC CURRENT

	Cost of Generating per net kw.-hr., 1923
Cost of operation:	
Operation	.01150
Maintenance	.00220
Allowance for maintenance	.00024
Depreciation	.00180
Interest on bonds	.00151
Amortization intangible assets	.00003
Bad debts	.00003
Itemized cost per kw.-hr., generating plant:	
Wages	.00208
Fuel	.00638
Lubricating oil	.00003
Miscellaneous	.00010
Maintenance	.00065

Milwaukee Engineers Hold Twentieth Anniversary

On the evening of Sept. 17, the Engineers' Society of Milwaukee commemorated its twentieth anniversary. Seven of the twenty past presidents participated, each reviewing his particular administration, pointing out the progress that had been made and urging the society to still greater activity. Messages of like import from some of the absentees were read and a fitting tribute paid to the two past-presidents who had passed away.

In 1904 the desire of the mechanical engineers in the city to form a local section did not meet with the approval of the national body. As a result the Engineers' Society of Milwaukee was formed. For a number of years it was made up entirely of mechanical engineers. In 1912, the local members of the A. I. E. E. formed a section which affiliated with the society, and this was the beginning of its enlarged activity. Later on the civil and refrigerating engineers did likewise, and the society grew until it now has an active membership of 300 and 750 affiliated members by virtue of their membership in local sections of the national societies previously mentioned.

In late years the society has been an important factor in civic affairs. Reference was made to the report on the Milwaukee River project, the assistance given the Harbor Commission, the work of the parking and traffic committee and the forming of a nucleus for a library on the applied sciences. All agreed that the society should continue its active participation in civic affairs. President Dorner anticipated an active season. The remodeling of the downtown district would give plenty of opportunity for work. No one was as well qualified as the engineer to render such service to the public and it was the common desire that work of this character should not only be continued but extended.

Senatore Luiggi Honored by Italy America Society

The Italy America Society held a luncheon in honor of the Honorable Luigi Luiggi, Italian engineer, at the Bankers Club, 120 Broadway, New York City, on Sept. 24.

Senatore Luiggi has just recently been appointed senatore, by the King of Italy, a position which is for life. He has been visiting in the United States for several weeks and has attended the celebrations of the Franklin Institute of Philadelphia and will be one of the speakers at the Centenary Celebration of the Rensselaer Polytechnic Institute at Troy, N. Y., to be held Oct. 3-4.

Paul Cravath is president of the Italy America Society, which has its rooms at 25 West 43 St., New York City.

Bituminous Coal Exports Drop During Summer

Total bituminous coal exports during August amounting to 1,392,862 long tons, showed a decline from shipments during July, when 1,630,849 tons were exported, but have been exceeded during 1924 only by June and July shipments, according to the Coal Division of the Department of Commerce. A large part of the decrease was due to smaller shipments to Canada, 1,049,346 tons being exported in August, against 1,202,400 tons in the preceding month.

Overseas exports of bituminous coal during the month in question were 335,671 tons, against 418,500 tons in July. A considerable part of this decrease was caused by smaller shipments to Italy, only 69,040 tons going to that country during August, against 115,055 in the previous month.

In the South American trade important decreases were witnessed in shipments to Brazil, the August exports amounting to only 80,892 tons, against 104,959 tons in July. There were no August shipments to Argentina or Chile, although exports to those two countries during July were 19,104 and 6,905 tons respectively. The total exports to South American destinations during August were 90,389 tons as compared with 139,098 tons in July.

Total shipments of coke amounted to 42,308 tons in August, against 48,983 tons in July, the decrease being largely due to smaller shipments to Canada, 34,953 tons in August, against 40,438 tons in the preceding month.

Exports of anthracite during August, amounting to 257,090 tons against 290,097 tons in July, were next to the smallest monthly shipments during the present year, the decline being mainly due to decreased shipments to Canada.

Water-Power Projects

Piru Creek, Calif., Project—An application for a preliminary permit has been filed with the Federal Power Commission by the Ventura Power Co. of Wilmington, Del. The application covers a project in Piru Creek, a tributary of the Santa Clara River in California. It is proposed to erect two dams, one 210 ft. and the other 200 ft. in height, so as to create reservoirs of 53,000 acre-feet and 22,000 acre-feet.

The Tirso, Sardinia, Hydro-electric Installation—The project will serve to irrigate about 75,000 acres of land in the Oristano plain and facilitate the reclaiming of a considerable area that is now waste land on account of periodic inundations, in addition to furnishing a large amount of electric power (estimated at 50,000,000 kw. hr. per year). This power will be available for industrial developments, especially the exploitation of the mineral resources of the island, which are more varied and extensive than those of any similar area in Italy. Moreover, the Tirso installation is merely the first of a long series of similar projects on the Tirso, Coghinas, Flumendosa and Terno Rivers, according to *Commerce Reports.*

Personal Mention

W. Herman Greul, consultant, announces the opening of his New York office in the Canadian Pacific Bldg., 342 Madison Ave.

Dr. Wilbur J. Huff, recently with the Koppers Co., Pittsburgh, has just been appointed professor of gas engineering at Johns Hopkins University.

F. E. Weymouth, chief engineer of the Bureau of Reclamation and an ardent advocate of the dam at Boulder Canyon, has recently resigned his position to go into private employment. Another case of an inadequate salary for government experts.

Society Affairs

The Engineers' Society of Western Pennsylvania has taken larger quarters in the William Penn Hotel, Pittsburgh, Pa.

The Lehigh Valley Section of the A. S. M. E. will hold a get-together meeting with a buffet supper on Oct. 3, at the Ascucon Valley Country Club.

The Western Washington Section of the A. S. M. E. will hold its Oct. 4 meeting at the Gowman Hotel, Seattle, when Ernest Hartford and Charles Penrose, of Philadelphia, will speak on topics of interest to the members of the section.

The Plant Engineers' Club of Boston, at a meeting of the executive committee, held recently, appointed the following members to serve on special committees: Program, W. S. Maddocks, K. D. Hamilton; power, Paul Cheever, H. A. Arey; membership, H. F. Scott. E. F. Burnham; service, H. T. Chandler, E. F. Brigham.

The Electrochemical Society, at its meeting in Detroit, will hold an all-day symposium on "Corrosion" on Oct. 2.

The New England Water Works Association, convention, to be held in Rochester, Oct. 1-3, will hear a paper by Chester M. Everett, consulting engineer, of New York City, on "The Diesel Engine Installation at New Britain."

The American Peat Society held a meeting at Minneapolis, Minn., Sept. 23, at which a study of the Minnesota peat lands was conducted. From the results of the investigations started the society hopes to make possible the reclamation of these barren wastes into valuable resources and mines of wealth. This study, of reclaiming and utilizing these deposits, begun a score of years ago, is to be continued.

The Engineers' Society of Milwaukee will hold its next regular annual session on Oct. 15. It will be a dinner meeting, and after routine matters are disposed of, short speeches will be given by members of the society who have recently returned from the World Power Conference. The speakers definitely scheduled to give their impressions of things abroad are: R. B. Williamson, electrical engineer of Allis-Chalmers Manufacturing Co.; T. C. Hatton, chief engineer of the Sewerage Commission of the City of Milwaukee; John Arthur Wilson, chief chemical engineer of A. F. Gallun & Co., Tannery. Other speakers that have been invited to speak are: John Anderson, chief engineer of the T. M. E. R. & L. Co.; William M. White, chief engineer and manager of the hydraulic department of the Allis-Chalmers Manufacturing Co.

Business Notes

The Economy Fuse & Manufacturing Co., Chicago, Ill., announces that its Minneapolis district sales office is removed to 1008 Marquette Ave., Minneapolis, Minn.

The Standard Water System Co., Hampton, N. J., announces some new agents to handle its power plant auxiliaries: Paulsen Spence Co., of New York City; H. W. Brandt, Philadelphia; Foster-Bolton, Inc., Chicago, Ill.

The Roller-Smith Co., maker of electrical instruments and circuit breakers, 233 Broadway, New York City, announces the appointment of the Thrall Electric Co., President Zayas No. 27 & Esquina A. Habana, Havana, Cuba, as its only representative for the island of Cuba.

Trade Catalogs

Insulation—Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. A new pamphlet, Circular 1686, has just been issued on the uses of "Micarta" in central stations. All the applications described are in use in some central station. Suggestions as to where micarta may be used to advantage are also shown. The circular is effectively illustrated.

Coming Conventions

American Association for the Advancement of Science. Burton E. Livingston, Smithsonian Institution Washington, D. C. Meeting at Washington Dec. 29-Jan. 3, 1925.

American Ceramic Society. Ross C. Purdy, Lord Hall, Ohio State University, Columbus Ohio. Convention at Los Angeles, Calif., Oct. 6-7.

American Electric Railway Association. James W. Welsh, 8 West 40th St., New York City. Meeting at Atlantic City, Oct. 6-10.

American Electrochemical Society. Dr. Colvin G. Fink, Columbia University, New York City. General meeting at Detroit, Mich., Oct. 2-4.

American Institute of Chemical Engineers. Dr. John C. Olsen, Polytechnic Institute, Brooklyn, N. Y. Annual meeting at Hotel Schenley, Pittsburgh, Pa., Dec. 3-6.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Fall convention at Pasadena, Calif., Oct. 13-17.

American Institute of Mining and Metallurgical Engineers. F. F. Sharpless, 29 West 39th St., New York City. Autumn meeting at Birmingham, Ala., Oct. 13-15.

American Society of Civil Engineers. 29 West 39th St., New York City. Fall meeting at Hotel Fuller, Detroit. Oct. 23-25.

American Society of Mechanical Engineers. Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 1-4.

American Society of Naval Engineers. Bryson Bruce. Bureau of Engineering, Washington, D. C. Annual meeting at Navy Bldg., Oct. 7.

American Society of Refrigerating Engineers. W. H. Ross, 35 Warren St., New York City. Annual meeting at New York City, Dec. 1-3.

American Welding Society. Miss M. M. Kelly, 29 West 39th St., New York City. Convention in Cleveland, Oct. 30-31.

Eastern Ice Manufacturers Association. W. H. Ross, 35 Warren St., New York City. Eighteenth annual meeting at Hotel Chelsea, Atlantic City, N. J., Nov. 12-14.

Electric Power Club. S. N. Clarkson, B. F. Keith Bldg., Cleveland, Ohio. Fall meeting at Greenbriar, White Sulphur Springs, W. Va., Oct. 20-23.

Empire State Gas and Electric Association. C. H. B. Chapin, Grand Central Terminal Bldg., New York City. Convention at Lake Placid Club, N. Y., Oct. 6-7.

Engineering Institute of Canada. Fraser S. Keith, 176 Mansfield St., Montreal, Que. Annual meeting at Montreal, Que., Jan. 27-29.

Illuminating Engineering Society. Norman D. MacDonald, 29 West 39th St., New York City. Convention at Briarcliff Lodge, Briarcliff Manor, N. Y., Oct. 27-31.

National Association of Practical Refrigerating Engineers. E. H. Fox, 5707 West Lake St., Chicago, Ill. Convention at New Orleans, La., Nov. 11-14. Exhibits at the Isaac Delgado High School.

National Exposition of Power and Mechanical Engineering. C. F. Roth, Grand Central Palace, New York City. Annual exposition at New York City, Grand Central Palace, Dec. 1-6.

National Safety Council, W. H. Cameron, 168 N. Michigan Ave., Chicago, Ill. Thirteenth Annual Safety Congress at Louisville, Ky., Sept. 29 to Oct. 3.

Society of Automotive Engineers. C. F. Clarkson, 29 West 39 St., New York City. Annual meeting at Detroit, Mich., Jan. 20-23.

Society of Naval Architects and Marine Engineers. Daniel H. Cox, 29 West 39th St., New York City. Annual meeting at 29 West 39th St., New York City, Nov. 13-14.

Instruments, Recording—The Foxboro Co., Inc., Neponset Ave., Foxboro, Mass. Leaflet describing a test given an 8-in. recording pressure gage.

Cement—Keystone Refractories Co., Inc., 120 Liberty St., New York City. Catalog "Dura-Stix All Temperatures Cement" describes the practical uses as a binder for firebrick, silica tile and granular refractories of this cement.

Heat Losses—Republic Flow Meters Co,, 2240 Diversey Ave., Chicago, Ill., A four-page pamphlet entitled "Heat Losses," by G. F. Gebhardt, Armour Institute of Technology, is article I of a series which this company is publishing on "Boiler Room Operation."

Compressors, Portable Air—Ingersoll-Rand Co., 11 Broadway, New York City. "One Hundred and One Ways to Save Money with Portable Air Power," is a comprehensive catalog covering the uses and varieties of portable air compressors. Illustrations and an index make it readily available for information on this subject.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Sept. 15, 1924	Sept. 22, 1924
Pool 1	New York	$2.75	$3.00
Smokeless	Columbus	2.00	2.25
Clearfield	Boston	1.90	2.35
Somerset	Boston	2.05	2.40
Kanawha	Columbus	1.40	1.55
Hocking	Columbus	1.55	1.65
Pittsburgh No. 8	Cleveland	1.85	1.90
Franklin, Ill.	Chicago	2.35	2.50
Central, Ill.	Chicago	2.20	2.25
Ind. 4th Vein	Chicago	2.35	2.50
West Ky.	Louisville	1.60	1.80
S. E. Ky.	Louisville	1.50	1.75
Big Seam	Birmingham	1.75	1.75

FUEL OIL

New York—Sept. 25, light oil, tank-car lots; 28@34 deg. Baumé, 4¾c. per gal.; 36@40 deg. 5c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Sept. 16, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.75 per bbl.; 26@28 deg., $1.80 per bbl.; 28@30 deg., $1.85 per bbl.; 30@32 deg., $1.90 per bbl.; 32@36 deg., gas oil, 5.5c. per gal.; 38@40 deg., 6c. per gal.

Pittsburgh—Sept. 9, f.o.b. local refinery; 30@34 deg., fuel oil, 5c. per gal.; 36@40 deg., fuel oil, 5¼c. per gal.

Dallas—Sept. 22, f.o.b. local refinery, 26@30 deg., $1.15 per bbl.

Philadelphia—Sept. 18, 28@30 deg., $1.99½@$2.06 per bbl.; 18@22 deg., $1.87½@$1.93¾; 13@16 deg., $1.75@$1.81½ per bbl.

Boston—Sept. 22, tank-car lots, f.o.b.; heavy oil, 12@14 deg. Baumé, 4.2c. per gal.; light oil, 28@32 deg. Baumé, 5¼c. per gal.

Cincinnati—Sept. 22, tank-car lots f.o.b. local refinery, 24@26 deg., Baumé, 4¾c. per gal.; 26@30 deg., 4½c. per gal.; 30@32 deg., 5c. per gal.

Chicago—Sept. 16, 20@22 deg., 5½c. per gal.; 24@26 deg., 6c. per gal.; 28@30 deg., 6c. per gal.

New Plant Construction

Calif., Los Angeles—Broadway Properties Co., Junior Orpheum Bldg., will take bids about Oct. 1st, for the construction of a 13-story theatre including steam heating and complete refrigeration systems at 838 South Broadway, estimated cost $1,000,000. G. A. Landsburgh, 700 Junior Orpheum Bldg., is architect.

Calif., Los Angeles—C. E. Garstang, 905 Trust & Savings Bldg., Archt., will soon award contract for the construction of a 5 story hospital, for the Protestant Methodist Church, estimated cost $500,000.

Calif., Los Angeles—The Engineering Dept. of the Municipal Power Bureau has plans underway for extending and improving the electric and power distributing systems. $16,000,000 bond issue was authorized at the election of Aug. 26th.

Calif., Merced—Merced Irrigation District, R. V. Meikle, Ch. Engr., Bancroft Bldg., will receive bids until Oct. 16, for power plant equipment as follows: 2 variable head turbines, 257 r.p.m. complete with governors; 2 generators, 15,620 kva., 11,450 v. 3 phase, 60 cycle, 257 r.p.m. with exciters and Kingsbury truss bearings; 1 switchboard complete with transformers, etc.; four 10,400 kva., 11,000 v. to 69,000 v. water cooled transformers; four 12,000 ampere 11,000 v. circuit breakers; two 55 cell storage batteries with 5 kw. motor generating set; 120 kva. lighting arrester.

Calif., Sacramento—The Supervisors of Sacramento County, H. W. Hall, Clk., is receiving bids for equipment for power house at the county hospital, estimated cost $55,000. R. A. Herrold, Forum Bldg., is architect for buildings.

Calif., San Francisco—C. V. Sachs, c/o S. Heiman, 57 Post St., Archt., is having preliminary plans prepared for the construction of an 8 story hotel on 17th Ave. and Geary St., estimated cost $500,000.

Del., Reedy Island—Treasury Department, Supervising Architect's Office, Washington, D. C., J. A. Wetmore, Acting Supervising Architect, will receive bids until October 16 for new water pumps, etc., at U. S. Quarantine Station, here.

Fla., Miami—Miami Realty Co., 108 Ave. B., North, L. W. Crow, Pres., plans the construction of a 15 story office building on North East 1st Ave., estimated cost $650,000. Hampton & Ehmann, architects.

Ill., Woodriver—City will vote about October 4 on proposition to issue $50,000 in bonds for water works improvements, to include 2 deep wells, 2 pumps and a Diesel engine.

Ia., Fort Madison—Artesian Ice Co., will build an addition to ice plant to bring capacity up to 100 tons of ice per day, electrically operated, estimated cost $75,000. H. Friedl, 501 North Dearborn St., Chicago, Ill., is architect.

Ky., Louisville—U. S. Engineers Office, has authorized the purchase of 3 boilers, locomotive type about 80 hp.

La., Chenevville—The Board of Aldermen and R. Deselle, Mayor, will receive bids until Oct. 20 for electric light, waterworks and ice systems, to include, 2 crude oil engines direct connected to alternators and clutch connected to 500 g.p.m. pump, complete 5 ton ice plant, etc. F. P. Joseph, engineer.

Md., Baltimore—The Consolidated Gas Electric Light and Power Co., Lexington Bldg., is preparing plans for the construction of a transformer station at 35th St., and Old York Road.

Md., Baltimore—Swindell Brothers, Bayard and Russell Sts., is in the market for two 40-hp. slip-ring motors, 220-440 volts, 25 cycle, 3 phase, 725 r.p.m., with starter and pulley, Westinghouse preferred.

Mass., Rutland—U. S. Veterans Bureau, Room 791, Arlington Bldg., Washington, D. C., F. T. Hines, Dir. will receive bids until Oct. 21st for the construction of various buildings including a boiler house here.

Mich., Detroit—R. W. Koch, 734 Book Bldg., Archt., is having plans prepared for the construction of a 12 story office building including a steam heating plant, estimated cost $700,000. Owner's name withheld.

Mo., St. Joseph—The City plans an election on November 18th, to decide whether to issue bonds for municipal light system or enter into contract to purchase power from private company, estimated cost $300,000.

Mo., St. Louis—I. T. Cook Co., Arcade Bldg., awarded a general contract for 5 additional stories to office building on Olive St. at 9th St., estimated cost $500,000.

Mo., St. Louis—Laclade Gas Light Co., Eleventh & Olive Sts., has filed application with the Missouri Public Service Commission at Jefferson City, for permission to expend $3,368,700 in plant improvements up to 1926. The plans include 2 new boilers, pumps, etc., in the proposed new gas holder at Station A, 148 Rutger St.

Mo., St. Louis—Washington University, Skinker & Lindell Sts., is having plans prepared for the construction of a power plant building and equipment, including 175 ft. brick stack, estimated cost $250,000.

N. Y., New York—A. M. Bing, 587 Fifth Ave., awarded the contract for the construction of a 15 story hotel on Lexington Ave. and 21st St.

N. Y., New York—J. David, 32nd St. and Broadway, will soon receive bids for the construction of a 20 story office building, estimated cost $1,500,000, Starrett & Van Vleck, 8 West 40th St., architects and engineers.

N. Y., New York—H. H. Jenkins, c/o Polhemus & Coffin, 15 East 40th St., Archt. and Engr., plans the construction of a 14 story loft at Madison Ave. and 37th St., estimated cost $2,000,000.

N. Y., New York—Papiders Eng. Corp., 210 West 70th St., will build an apartment hotel, at 27 West 73rd St., estimated cost $3,000,000. Work to be done by separate contracts.

N. Y., New York—910 West End Ave. Corp., c/o G. F. Pelman, 200 West 72nd St., Archt. and Engr., plans the construction of a 15 story apartment on West End Ave. and 105th St.

N. Y., New York—322 West 72nd St. Corp., M. M. Silverman, Pres., 347 Madison Ave., had plans prepared for the construction of a 15 story apartment house, estimated cost $1,000,000. G. & E. Blum, 505 Fifth Ave., architect and engineer.

N. Y., New York—J. N. Trainer, c/o T. W. Lamb, 644 Eighth Ave., Archt., plans the construction of a theater and hotel, on 42nd St. and 8th Ave., estimated cost $1,500,000.

N. Y., Woodside—(Flushing P. O.)—United Triangle Corp., c/o E. DeRosa, 110 West 40th St., Archt., is having plans prepared for the construction of a theater, apartment and bank building on Woodside Ave. between 59th and 60th Sts., estimated cost $1,000,000.

O., Lorain—Syndicate, c/o J. H. Drummond, 450 Euclid Ave., Cleveland awarded the contract for the construction of a 4 story, stores and office buildings, corner of Broadway and East Erie Ave., estimated cost $500,000. E. McGeorge, 3030 Euclid Ave., architect.

Okla., Fort Sill—Office of the Constructing Quartermaster will receive bids until Oct. 18th, for the construction of 3 boiler houses and heating plants for buildings, 113, 115 and 123.

Pa., Bristol—W. J. Steel, awarded a contract for a boiler house 40 x 50 ft. cost about $20,000. The Austin Co., B & Clearfield Sts., Philadelphia, are engineers and builders.

Pa., Philadelphia—Quartermasters Intermediate Depot, 21st St. & Oregon Ave., will receive bids until Oct. 13th, for construction of an auxiliary boiler house and boiler installation, also additional fuel oil burning equipment in 2 boilers.

Pa., Philadelphia—Union Tank Car Co., 21 East 40th St., New York, N. Y., awarded the contract for the construction of a power house on Penrose Ave., estimated cost $50,000.

Pa., York—Community Hotel Co., c/o C. P. Rice, Pres., will soon award contract for the construction of a 10 story hotel, W. L. Stoddart, 50 East 41st St., architect and engineer.

Tenn., Memphis—Tietig & Lee, 901 4th National Bank Bldg., Cincinnati, Ohio, Archts., are having plans prepared for the construction of an 8 story office building including 3 elevators, ice plant, etc., on Madison and 4th Sts., for the Medical Arts Building.

Tex., Greenville—City plans a water works system including pumping equipment, estimated cost $350,000. Elrod Engineering Co., Interurban Bldg., Dallas is engineer.

Tex., Lampasas—Lampasas Ice & Refrigerating Co. plans the construction of a refrigerating plant with a cold storage capacity of 14,000 cu.ft., estimated cost $35,000.

Tex., Port Arthur—City, J. P. Logan, Mayor is in the market for a 20,000 gallon pump equipped with motor.

Tex., San Antonio—R. H. Cameron, City National Bank Bldg. Archt., will soon award the contract for a 12 story medical arts building on Ave. E, & Plaza St., estimated cost $1,000,000. Owner's name withheld.

Tex., Vernon—Vernon Texas Power & Light Co., c/o Morrison & McCall, Inc., will soon receive bids for the construction on an ice plant, estimated cost $42,000. Private plans.

Wash., Kalama—The Port of Kalama has filed application with M. Chase, Supervisor of Hydraulics, for appropriation of 500 sec.ft. of water from the Kalama River, and permission to construct a reservoir to impound 500 acre ft. It is proposed to convey water by conduit 7¼ miles, and to develop 11,350 hp., estimated cost is $1,100,000.

Wash., Seattle—The City Council finance committee has approved an ordinance providing for immediate further development of the Skagit power project. Applications for permit will be made for water power rights on the Skagit River at Ruby & Seattle Creeks, about 10 miles from the present Gorge plant. Bonds will be issued for the construction of a dam at Ruby Creek and a power plant at Seattle Creek, estimated cost of dam is about $8,500,000. C. F. Uhden, is chief engineer for Skagit Project.

W. Va., Wheeling—Wheeling News Lithograph Co., is in the market for a gas engine generator unit, 500 kw. capacity or more.

Wis., Gays Mills—The Inter State Power Co., McGregor, Iowa, are planning the construction of a dam here, and additions and improvements to power plants at Soldiers Grove and Bloomington.

Wis., Milwaukee—Corcoron College, 3127 State St., awarded the contract for the foundation of a power house on 31st and State Sts., estimated cost $40,000.

Wis., Milwaukee—Cudahy Family Co., J. Bannen, Mgr., 85 Michigan St., plans the construction of a 6- or 8-story office building on Milwaukee Ave., estimated cost $1,000,000. Architect not selected.

Wis., Plymouth—Plymouth Refrigerating Co., E. J. Larson, Mgr., plans the construction of a cold storage plant, estimated cost $150,000. E. A. Juul, Security Bldg., Sheboygan, architect.

Wis., Sheboygan—Midwest Fibre Co., A. Dedelack, Pres., is having plans prepared for a steam heating boiler plant in factory building. Boilers, etc., will be required. W. C. Works, 720 Ontario Ave., engineer.

N. B., St. Stephen—Martime Electric Co., plans to rebuild plant destroyed by fire, estimated cost $250,000.

N. Z., Wellington—Public Works Tender Board will receive bids until November 18th, for Wailkats electric power supply, section 103; two oil circuit breakers, panel, excluding kva. meter, potential transformers, current transformers; 21 isolating switches, 6,600 v. and 5,000 v. lightning arresters, kva. meter.